

FOURTH EDITION

A WORKING GUIDE TO Process Equipment



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Norman P. Lieberman
Elizabeth T. Lieberman

A Working Guide to Process Equipment

Norman P. Lieberman
Elizabeth T. Lieberman

Fourth Edition



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To the union of two people
Weathering life's storms together
Watching the lightning
Waiting for the thunder
In friendship, In partnership
In love

To the Memory of
Our Friend and Colleague

Gilles de Saint Seine
Process Engineer
Total-Fina-Elf, France

It's more than losing a friend, it seems as if
Liz and I have lost part of ourselves, but we
will always remember his gentle determination
and insightful work, his love of family and
consideration for his colleagues, and not least
his marvelous wit.

This book is dedicated to our parents:

Elizabeth and Tom Holmes, innovative engineers,
courageous under fire at war and in peace.

Mary and Lou Lieberman whose enduring strength and
fortitude have been little noted, but long remembered.

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Foreword

The forest has largely reclaimed the Amoco Refinery in Destrehan, Louisiana. A large sugar maple sprouts from a long since forgotten control room. If one didn't know, the delayed coker drum structure might be mistaken for a ruined temple. Brick chimney stacks rise defiantly above the feathery tops of the cypress trees.

The process vessels were uprooted in 1955 and shipped to the Amoco Refinery in Texas City. Odd bits of piping and valves have sunk into the swamp. A half century of gentle, but persistent effort by Mother Nature has partly reclaimed this bit of earth for the natural world.

The area is fenced and closed to the public. Well test points dot the fence line. After 50 years, someone is still monitoring the ground water seepage for escaping aromatics and other cancer-causing chemicals. I've hiked through this area a dozen times. Never have I seen a rabbit or a snake or an alligator or a nutria or a deer. The large pond in the center of the site is devoid of fish, turtles, and frogs.

Some day soon—in 20 years or 100 years—the Mississippi River, which flows just across the River Road, will breach its levee and clean up this mess. Or nature may get really angry and toss the Gulf of Mexico over Destrehan with a category 5 hurricane. It's only a matter of time.

Same with the other problem. CO₂ is increasing at 2 ppm a year. Another 3 or 4°F will make large parts of the earth uninhabitable. Exploitation of heavy hydrocarbon deposits in Alberta and Venezuela can only increase the 2 ppm CO₂ growth rate. This can go on for another 20 years or another 100 years. Then Mother Nature will take serious corrective action to restore natural equilibrium. It's only a matter of time.

What can you do? A few ideas:

- Close hand valves on steam turbines
- Reduce pump impeller sizes
- Suction throttle compressors rather than spillback recycle gas

- Minimize tower pressure to maximize relative volatility and minimize reflux rates
- Keep gas-fired turbine rotors clean
- Avoid afterburn in heater convective sections
- Don't justify new projects to replace existing equipment that just needs maintenance
- Read this book

Sooner, rather than later, nature may get really angry and eliminate the fundamental problem. Time is not on our side.

Other Books by Norman P. Lieberman

- *Troubleshooting Refinery Processes* (1980 edition)
- *Troubleshooting Natural Gas Processing*
- *Process Design for Reliable Operations* (3rd edition)
- *Troubleshooting Process Plant Control*
- *Process Engineering for a Small Planet*
- *Process Equipment Malfunctions*
- *Troubleshooting Process Operations* (4th edition)
- *Troubleshooting Vacuum Systems*
- *My Race Against Death: The Story of One Runner Who's Running Further and Faster with Age, But Who Refuses to Listen to Reason*

The best method to purchase any of these texts is Amazon. *A Working Guide to Process Equipment* is the most popular of the above list. *Troubleshooting Process Operations* is the best text for refinery-specific applications. Young engineers and operators find *Troubleshooting Process Plant Control* the most helpful. Check our website for details: www.lieberman-eng.com.

Preface to the Fourth Edition

Buried deep in our genetic heritage, hidden in an obscure string of DNA, is coding for Process Equipment Operations. An instinctive desire to apply energy to transmute the properties of naturally occurring materials into other, more useful forms. Like cooking food; or fusing clay into ceramic pots; or reacting sulfur with air to produce sulfuric acid; or transmuting lead to gold.

Looking back on 50 years as a process engineer, the most satisfying period of my career was devoted to converting gas oil into viscous polypropylene via cracking reactions and refrigeration.

The gene that codes for operation and design of process equipment, such as distillation columns and fired heaters, is a recessive gene. Only one out of 40 individuals have inherited this genetic code for process equipment operations as a dominant trait.

Should you wish to determine if your child has inherited this genetic makeup for process equipment operations, observe if the child:

- Is fascinated by fire
- Tries to dam and divert little streams
- Is attracted by boiling water
- Asks what makes a windmill turn

Thus, only one out of 40 people have the potential to evolve into process engineers or operators. The rest will become Directors of Human Resources or Maintenance Superintendents.

My older sister often asks, "Norman. You're over 72. When are you going to retire? You're too old to be climbing distillation towers. You'll fall off one of these days."

"Arline," I explain, "I can't retire. It's in my blood."

XX Preface to the Fourth Edition

“Norman, you’re crazy! Everyone else in our family retired in their 60s. It couldn’t be in your blood. Dad moved to a retirement village when he was only 62.”

“You don’t understand, Arline. It’s a recessive gene I inherited from our ancestors generations ago. I can’t retire. It’s part of my DNA. It’s instinctive behavior. Like a beaver building a dam. Or squirrels gathering nuts in the fall. I can’t retire. I’ll just have to go on until the end.”

Norm Lieberman

If you have questions, you can contact us at:

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Preface to the Third Edition

When I feel a gentle breeze cool against my skin, I think about the sun causing uneven heating of the earth's surface. Hot air rises; cool air rushes into the area of lower air density. Thus the wind.

When I see a sparkling stream cascading in silver bubbles over rounded pebbles, my thoughts drift to converting potential energy to kinetic energy. Our visit to the lovely wine country of the Sonoma Valley awoke within me memories of the early days of spirit distillation.

A wax candle was burning in a remote church we visited in England. How many candles would be burned to heat the air in the church by 10°F? Knowing the heat of combustion of wax and the specific heat of air, I could and did calculate that 68 candles would be consumed.

A child's brightly colored pinwheel spinning in the wind creates an irresistible urge to discourse on steam turbines. A pot of pasta boiling over on our stove is the perfect incentive to deliver a lecture, which no one wants to hear, detailing foam-induced flooding in packed towers. Suntanned skin is the perfect example illustrating the power of radiant heat transfer.

I quite realize that normal people do not have such thoughts racing continuously through their minds. I can't help myself. It's a consequence of being a process engineer for too long.

For 43 years I've been worrying and pondering about how process equipment works. For half a century I've been seeking enlightenment about mysteries of nature!

- What causes thermosyphon circulation in reboilers?
- How can a positive pressure develop in a natural draft-fired heater?

- What causes surge in centrifugal compressors?
- Why do two identical condensers, working in parallel, perform entirely differently?
- What are the hand valves on a steam turbine all about?
- How exactly does reflux improve product separation?
- Why is the seal flush pressure to a centrifugal pump the suction rather than the discharge pressure?
- What affects tray efficiency in distillation?
- Why do reciprocating compressors have adjustable unloaders?
- How can the liquid level in a drum of boiling water be measured?
- Can the oil temperature inside a pipe be measured based on the exterior pipe temperature?
- Why do some heat exchangers suddenly seem to clean themselves?

Maybe if Liz and I write all this down, I can stop thinking about these questions. We tried to do so in our first two editions, but failed. I'm still wondering and worrying. So once again we will try in this third edition of *A Working Guide to Process Equipment: How Process Equipment Works*. If you have questions, contact us at:

- 1-504-887-7714 (phone)
- 1-504-456-1835 (fax)
- Norm@Lieberman-eng.com (email)

Norm and Liz Lieberman
Metairie, Louisiana
May 7, 2008

Preface to the Second Edition

Dear Aunt Hilda:

I hope this letter finds you well and in good spirits. How is Uncle Herb?

Incidentally, I am not a drug dealer. My mother, I fear, has told you that I make a lot of money in drugs. I sometimes consult on process engineering problems for pharmaceutical companies—but I do not deal drugs.

I know my mother told you about the drugs because she doesn't understand what Liz and I do. Also, she's still disappointed that I didn't become a doctor. I've explained what a process engineer does to Mom a hundred times. Let me try to explain it to you.

First, Aunt Hilda, process equipment is all around us:

- Gas burner in an oven
- Steam radiator
- Vacuum cleaner
- Sump pump in your basement
- Central air conditioning
- Hot water heater
- Toilet water closet
- Refrigerator

Our job is to design improvements to existing process equipment. To do this we do three things:

- Investigate the current operation of the process equipment in oil refineries and chemical plants. Based on field data, what are the actual operating parameters of a process plant?

- What are the design or theoretical operating parameters of the plant?
- What is causing the plant to perform below its theoretical efficiency and what shall be done to correct the deficiency?

To be successful, Liz and I have to understand in a fundamental way how the equipment actually works. It's true that I do a lot of work with centrifugal refrigeration compressors. However, contrary to what my mother told Mrs. Goldberg, I am not a Frigidaire repairman.

I hope this letter explains what I've been doing for the past 40 years. I definitely do not regret not becoming a doctor. I'm enclosing a copy of our book, *A Working Guide to Process Equipment: How Process Equipment Works*, for you and Uncle Herb. This really explains what process equipment is all about in a pretty simple way.

I think you would agree that an important company like McGraw-Hill would not publish such a book if they thought that I was a drug dealer. We'll see you, God willing, at the wedding in June.

All our love,

Norm and Liz Lieberman

January 4, 2003

P.S.: If you have any questions, please contact us at:

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Preface to the First Edition

November 1, 1996

Subject: How Process Equipment Works

Dear Reader:

Thank you for buying our book. We worked very hard writing it, and we appreciate your vote of confidence.

No normal person is going to read this book for fun or relaxation. It is a work book, for working people. You purchased it with the hope and faith that it can help you do a better job. You opened it with the expectation that you can read it with comprehension.

Well, we won't let you down. But, let's make a deal. We promise you that even though this is a technical book, you can read it easily, without pain, but with comprehension. After you read it, you will definitely be a better process operator or engineer. Your part of the deal is to read the whole book. This is not a reference or source book. All the chapters are tied together by threads of logic. You will really find it easier to grasp this logic if you read the chapters in sequence.

A few of the words in the text are italicized. These words are explained in the Glossary at the back of the book.

Please feel free to give us a call if there is some point you would like to discuss, or a process question you wish to ask.

Sincerely,

Norm Lieberman
Chemical Engineer

Liz Lieberman
Chemical Engineer

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Introduction

In 1983, I started teaching a three-day process equipment troubleshooting seminar to chemical engineers and experienced plant operators in the petroleum refining and chemical process industry. Since the inception of the seminar, in excess of 7000 men and women have attended the classes. The seminar is largely based on my 40 years' experience in field troubleshooting and process unit revamp design.

I have taught hundreds of seminars explaining how pumps, compressors, heat exchangers, distillation towers, steam jets, fired heaters, and steam turbines malfunction. I have explained to thousands of chemical engineers how to design trays and modify tube bundles for improved performance. More thousands of operators have listened to me expound as to how and why cavitation damages pump mechanical seals. And throughout these lectures, one common thread has emerged.

The general knowledge as to how process equipment really functions is disappearing from the process industries. This is not only my opinion but the general view of senior technical managers in many large corporations.

Chemical process equipment is basically the same now as it was in the 1930s. The trays, K.O. drums, compressors, heaters, steam systems have not changed—and probably will not change. The fundamental nature of process equipment operation has been well established for a very long time. Modern methods of computer control and process design have not, and cannot, change the basic performance of the bulk of process equipment. These tools just have made learning about the working of the equipment more difficult.

The chemical engineer has traditionally been the guardian of process knowledge. So, one would suppose that if fundamental process knowledge is vanishing, the origin of the problem may lie in our universities. Perhaps there is less of that “hands-on approach”

to problems with the advent of the PC or perhaps there are just fewer people around to teach us. No one really knows.

But in this book, we have gone back to the very simplest basis for understanding process equipment. In every chapter we have said, “Here is how the equipment behaves in the field, and this is why.” We have shown how to do simple technical calculations. The guiding idea of our book is that it is better to have a working knowledge of a few simple ideas than a superficial knowledge of many complex theoretical subjects.

The original three-day troubleshooting seminar has now grown into a six-day course that only covers about 50 percent of the subjects I tackled in the three-day class. Why? Because it is no longer primarily a troubleshooting seminar. The vast majority of the class time is now devoted to explaining how the equipment really functions and answering the following sorts of questions:

- Why do trays weep?
- Why do weeping trays have a low tray efficiency?
- What does tray efficiency actually mean anyway?
- Is there a way to design trays that do not weep?
- Why should an operator need to know why trays weep?
- Can a tray weep, even though the computer calculation says the tray cannot weep?

Several years ago I began to make notes of the questions most frequently asked by my clients and students. Sometimes, it seems as if I have been asked and have responded to every conceivable process equipment question that could possibly be asked. Certainly, I have had plenty of practice in forming my answers, so that they are comprehensible to most process personnel, maintenance people, and even management. We have tried to summarize these questions and answers in this book.

Like everybody else, I have answered questions without always being correct. But over the years, I have continued to learn. I have been taught by the source of all wisdom and knowledge—the process equipment itself. I am still learning. So you could say that this book is a progress report of what I have learned so far. I think that my troubleshooting field work and revamp designs have acted as a filter. This filter has removed, and still is removing, from my store of knowledge misconceptions as to the true nature of process equipment functions.

You do not need a technical degree to read and understand this text. Certainly this is a technical book. But the math and science discussed is high school math and science. We have traded precision for simplicity in crafting this book.

Liz and I would be happy to discuss any questions you might have pertaining to the process equipment discussed in our book. You can phone or fax us in the United States at:

Phone: (504) 887-7714

FAX: (504) 456-1835

But if you call us with a question, my first response is likely to be, "Have you looked the problem over in the field?"

Norman P. Lieberman

Email: norm@lieberman-eng.com

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Acknowledgments

Fortunately for us there are those who benevolently accept that students, just like children, are always with you.

The authors gratefully acknowledge the most kind and generous support of J. F. Richardson (JFR), in the production of the second edition, and also the timely assistance provided by R. P. Chhabra (Raj), J. Hearle (Jonathan), and J. R. Condor. The areas of this book dealing with fluid flow in pipes and hazardous pipe work would have been impoverished without their help and guidance.

My days at the University College Swansea (South Wales, UK), both during undergraduate years and at times since then when I have had the need or opportunity to revisit, are very special to me. The college and its staff have provided a rock-steady foundation for everything that comes my way as an itinerant Chemical Engineer alongside my coauthor Norman.

I will never forget the experience of working (quite literally) in the field with JFR, Raj, and Jonathan: sometimes paddling through inches of china clay slurry in an isolated pump house, or tracking down vent valves for pressure readings on a cross-country pipeline that traversed several miles in the heart of Cornwall (UK).

That hands-on training and field work, just as it should be, was very much part of the academic work of the thriving Chemical Engineering Department at the University College Swansea. We have to thank late Prof. J. F. Richardson for that, and we hope that his concept will live on and grow as a model to us all in his absence.

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CHAPTER 1

Process Equipment Fundamentals

I couldn't help but notice that the blue fish was permanently dead. Most sadly, it was floating on its side. The cause of death was clear. The water circulation through the aquarium filter had slowed to a thin trickle. Both the red and silvery striped fish also appeared ill. I cleaned the filter, but the water flow failed to increase.

As you can see from Fig. 1.1, the filter is elevated above the water level in the fish tank. Water is lifted up, out of the tank, and into the elevated filter. Water flowing up through the riser tube is filtered, and then the clean water flows back into the aquarium.

I tried increasing the air flow just a bit to the riser tube. The water began to gurgle and gush happily through the filter. Encouraged, I increased the air flow a little more, and the gush diminished back to a sad trickle.

It was too bad about the blue fish. It was too bad that I didn't understand about the air, or the filter, or the water flow. It was really bad because I have a master's degree in chemical engineering. It was bad because I was the technical manager of the process division of the Good Hope Refinery in Louisiana. Mostly, it was bad because I had been designing process equipment for 16 years, and didn't understand how water circulated through my son's aquarium.

Maybe they had taught this at my university, and I had been absent the day the subject was covered? Actually, it wouldn't have mattered. Absent or present, it would be the same. If Professor Peterson had covered the subject, I would not have understood it, or I would have forgotten it, or both. After all, "Universities are great storehouses of knowledge. Freshmen enter the university knowing a little, and leave knowing nothing. Thus, knowledge remains behind and accumulates."

But then I realized that I had seen all this before. Six years before, in 1974, I had been the operating superintendent of a sulfuric acid regeneration plant in Texas City. Acid was lifted out of our mix tank by injecting nitrogen into the bottom of a 2-inch riser pipe. The shift operators called it an "air lift pump."

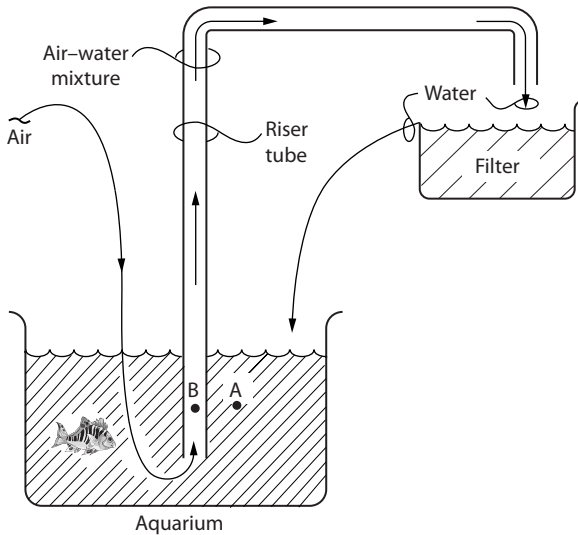


FIGURE 1.1 An air lift pump circulates water.

The problem was that in 1974 I didn't understand how the acid air lift pump worked either. More nitrogen pumped more acid. That's all we knew in Texas City, and all we cared to know.

Thinking about Texas City and my university days, my thoughts drifted to an earlier time. Back before my high school days in Brooklyn. Back to my childhood and to memories of my yellow balloon. The balloon was full of helium and I lost it. The balloon escaped because it was lighter than air. It floated up, up, and away because the helium inside the balloon was less dense than air. The yellow balloon was lifted into the sky because of the density difference between the low molecular weight helium inside the balloon, and the higher molecular weight of the surrounding sea of air.

So that's what makes an air lift pump work; density difference. Density difference between the lighter air-water mixture in the riser tube and the more dense water in the fish tank.

In Fig. 1.1, the pressure at point A will be greater than the pressure at point B. It's true that the height of liquid in the riser tube is triple the height of water in the tank. But because of the bubbles of air in the riser tube, the density of the mixed phase fluid in the riser is small compared to the density of water. The pressure difference between point A and point B is called the "airlift pump driving force." Water flows from an area of higher hydrostatic head pressure (at A) to an area of less hydrostatic head pressure (at B). Using more air reduces the density in the riser tube. This lowers the pressure at point B. The differential pressure between A and B increases.

The greater driving force then increases water flow through the aquarium's filter.

This still leaves a problem. Why did the second increment of air flow reduce the rate of water circulation?

1.1 Frictional Losses

We used to make wooden knives in Brooklyn, New York, by rubbing a stick on the sidewalk. The wood never got too sharp, but it did get hot. Sometimes it even smelled smoky when I rubbed the wood fast enough. More speed, more friction. Friction makes heat.

When the air–water mixture flows up through the riser tube, the potential energy (meaning the height of the circulating water) increases. The energy to supply this extra potential energy comes from the pressure difference between point A and point B. Some of the air lift pump driving force is converted into potential energy.

Unfortunately, some of the airlift pump driving force is also converted to frictional losses. The friction is caused by the speed of the air–water mixture racing up through the riser tube. More air means more flow and greater velocities, which means more friction. Too much air makes too much friction, which means less of the air lift pump driving force is left for increasing the potential energy of the water flowing up into the filter. At some point, increasing the air flow reduces water flow up the riser due to an increased riser tube pressure drop because of friction.

1.2 Density Difference Induces Flow

I'd better phone Professor Peterson to apologize. I just now remembered that we did learn about this concept that density difference between two columns of fluid causes flow. Professor Peterson taught us the idea in the context of draft in a fired heater. Cold combustion air flows through the burners and is heated by the burning fuel. The hot flue gas flows up the stack. The difference in density between the less dense hot flue gas and the more dense cold air creates a pressure imbalance called draft. Just like the fish tank story.

However, I can't call Professor Peterson. He's dead. I wouldn't call him anyway. I know what he would say: "Lieberman, the analogy between the air lift pump and draft in a fired heater is obvious to the perceptive mind, which apparently excludes you."

1.3 Natural Thermosyphon Circulation

I worked as a process design engineer for Amoco Oil in Chicago until 1980. Likely, I designed about 50 distillation columns, 90 percent of which had horizontal, natural thermosyphon circulation reboilers.

I saw hundreds of such reboilers in Amoco's many refineries. I never stopped to think what caused the liquid to circulate through the reboilers. I never thought about it, even though the reboiler feed nozzle on the tower was below the vapor return nozzle. Now, with my fish tank experience as a guide, I was able to understand:

- The reboiler shell is like the fish tank.
- The reboiler vapor is like the air.
- The reboiler return pipe is like the riser tube.
- The distillation tower is like the filter.

Every Saturday I run for 6 miles along the levee bordering the Mississippi River in New Orleans. Huge sand hills lie between the levee and the river. The sand has been dredged from the river bed by the Army Corps of Engineers. The Corps uses 30-inch diameter flexible hoses to suck the sand from the river bed. Maybe the concept of "sand sucking" is not the most elegant terminology? To be precise, a barge floating on the river, equipped with an air compressor, discharges air to the bottom of the 30-inch hose, 140 ft below the surface. The reduced density inside the hose, due to the compressed air, creates an area of low pressure at the bottom of the hose. The water and sand are then drawn into the area of low pressure and up the hose, which empties the sand and water into a basin along the riverbank. You can see a geyser of water and sand spurting up in these sand basins. I made a mini-dredge like that to suck the sand out of my pool sand filter. It worked rather well, until the little air compressor motor began smoking.

1.4 Reducing Hydrocarbon Partial Pressure

One day my mother served me a bowl of mushroom soup which I didn't want to eat. I disliked mushroom soup, but I was a practical child. It would serve no purpose to tell my mother I hated the taste of mushrooms because she would say, "I've spent all day cooking. You're not going outside till you eat that soup." So I said, "Mom, the soup is too hot. I'll burn my tongue." And she said, "Norman, blow across the soup to cool it off." While I knew this would cool the soup, I really didn't like mushrooms. So I responded, "Mom, why will blowing across the soup cool it off? How does that work?"

At this point your typical mother would slap the kid in the head and say, "Children in Europe are starving (this was in 1947; now European children are overweight). Shut up and eat your soup." But not my mother. "Norman, blowing across the soup blows away the molecules of steam covering the top of the soup. This makes room for more molecules of water to escape from the surface of the soup in the form of steam. When the molecules of water are changed into molecules

of steam, that takes a lot of heat. This heat is called latent heat. This latent heat does not come from your breath, which is colder than the soup. The heat to vaporize the soup comes from the hot soup itself. The temperature of the soup is called sensible heat. When you blow across the soup, you're helping the sensible heat content of the soup to be converted to latent heat of evaporation of the soup. And that's why the soup cools. But your breath simply acts as a carrier—to carry away the molecules of steam covering the surface of the soup."

And I said, "What?"

And Mom said, "Norman, in effect, your breath is reducing the partial pressure of steam in contact with the soup. For every one weight percent of evaporation, the soup will cool by 10°F."

If my mother had served me a hydrocarbon soup, then for every one weight percent of evaporation, the soup would have cooled by 2°F. Then she would have said the carrier gas or stripping steam would be reducing the hydrocarbon partial pressure.

I have designed process equipment where the carrier medium is the air. Sometimes we use nitrogen or hydrogen. But mainly we use steam because it's cheap and condensable. We use steam:

- in the feed to towers.
- as the stripping medium in steam strippers.
- in evaporators.

The steam is used to promote vaporization of the product. But the heat of vaporization does not come from the steam, it comes mainly from the product itself. This is true even if the steam is superheated.

As an adult, I grow my own mushrooms on logs and consume them quite happily. Mom's gone now, and I would give a lot for a bowl of her mushroom soup. But I still remember the lesson about the reduction in partial pressure and the conversion of sensible heat to latent heat.

1.5 Corrosion at Home

My mother always thought that I was a genius. She would tell all the other mothers in our neighborhood, "You should have your daughter meet my son, he's a genius." My mother decided that I was a genius based on one incident that happened when I was six years old. She called me into the bathroom. "Norman! Look at the sink." The sink was discolored by brown, rusty stains from the old pipes in our ancient apartment house.

"Mom, I think my sister did that. It's not my fault. It's Arlene's fault."

"Norman, no one is blaming you for the stains. Stop blaming Arlene. What I want is for you to get the stains off."

So I went into the kitchen, got a bottle of Coke, poured it over the stains, and the sink became clean. From this single incident, my mother

decided I was a genius and that all the teenage girls in south Brooklyn should fall in love with me. Actually, I went out with one of those girls—Gloria Harris. I really liked her. But she dumped me. Gloria told her mother that I was just another nerd.

What was it about the Coke that removed the iron deposits from our sink? It was carbonic acid (H_2CO_3). (Coke contains lots of phosphoric and citric acid too.)

Carbonic acid is formed when CO_2 dissolves under pressure in water. The resulting acid has a 5 to 6 pH, even at relatively high acidic concentrations. The acid readily dissolves iron to form water-soluble iron carbonate, $\text{Fe}(\text{HCO}_3)$.

This is a problem in process plant steam heaters. There are always some residual carbonates in boiler feed water. When the water is turned into steam, some of these carbonates decompose into CO_2 . Thus, all steam is contaminated with CO_2 . The CO_2 being far more volatile than water gets trapped and accumulates in the high points of steam heaters. With time, the CO_2 condenses in the water to form carbonic acid. This causes corrosion and tube leaks. To avoid CO_2 accumulation, the exchanger high points can be vented.

I knew all this when I was a child. Not the carbonic acid part. I knew that Coke dissolved rust stains from sinks. I had seen Mrs. Fredirico, my friend Armand's mother, clean a sink with Coke so I knew it would work. That's my idea of applied technology—applying the experiences of ordinary life to process problems. I tried to explain this to Gloria, but we were both teenagers and she wasn't interested. If she knew how much money I've made from my childhood experiments, I bet she would be sorry now.

1.6 What I Know

Sometimes I work with process equipment as a field troubleshooter. Sometimes I specify equipment as a process design engineer. And often, I teach shift operators and plant engineers how equipment works. Whatever I'm doing, I have in mind my childhood experiences in south Brooklyn. I focus on the analogy between the complex problem of today and the simple experiences of everyday life.

I often have my head in the clouds, but I always keep my feet on the ground. I learned this from my mother. She was a great storehouse of knowledge. And I've continued to learn as an adult too. Let me explain.

1.6.1 Toilet Training

The first skill that a new homeowner should acquire is toilet repair. I had my first lesson on this vital skill in 1969. We had just moved into our first house in south Chicago when I discovered our toilet wouldn't flush. An experienced co-worker at the American Oil Refinery in Whiting, Indiana (now BP), suggested that I check the roof vent (see Fig. 1.2).

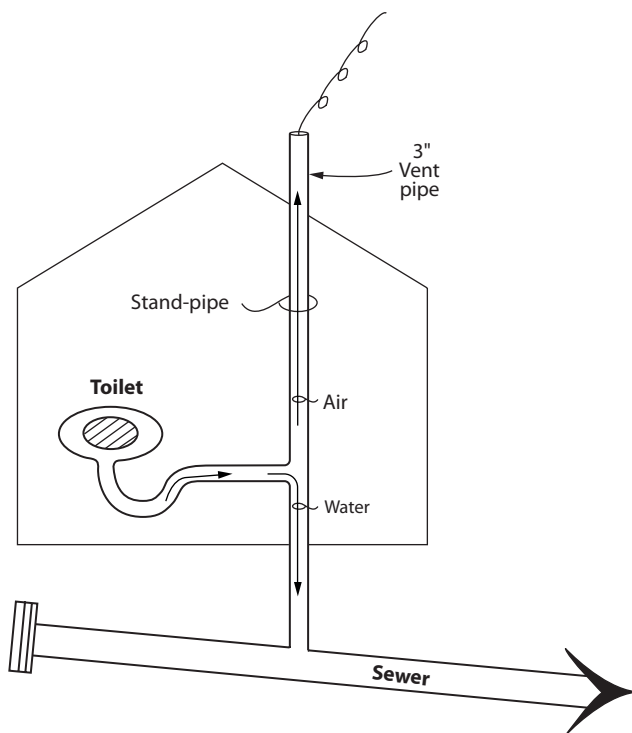


FIGURE 1.2 My toilet roof vent.

Climbing onto the roof I found that a pigeon had built its nest on top of the 3-inch diameter vent pipe. I removed the nest and the toilet flushed just fine. The water swirled around merrily in the bowl for a few seconds. Next, the water gushed and rushed down the toilet's drain with wonderful speed and vitality. The water seemed to be in such a hurry to leave the toilet bowl and escape through the sewer that it dragged a small amount of air with it.

The verb "to drag" is a poor engineering term. The correct technical terminology describing this well-known phenomenon is that the rushing water sucked the air down the toilet's drain. But the sucking of air out of my bathroom could only happen if the pressure in the toilet's drain was less than the pressure in my bathroom. This idea bothered me for two reasons:

1. What caused a sub-atmospheric pressure (a partial vacuum) to develop at the bottom of my toilet bowl?
2. Where did the air sucked down into the drain go to?

Here's the way it seems to me: When we flush the toilet, the velocity or the kinetic energy of the water swirling down the bowl increases.

The source of this kinetic energy is the height of water in the water closet. That is the potential energy of the water. We're converting potential energy to kinetic energy in accord with Bernoulli's equation.

If you live in an apartment house in Brooklyn, there is no water closet. The water supply for the toilet comes directly from the high pressure water supply line. Then we are converting the water's pressure to the velocity of water rushing into the toilet bowl. Either way, the spinning, draining water develops so much kinetic energy that the pressure of the water falls below atmospheric pressure. A slight vacuum is formed, which draws a small amount of air down the toilet's drain.

When the air-water mixture enters the larger, vertical stand-pipe in Fig. 1.2, the velocity of the air-water mixture goes down. Some of this reduced kinetic energy is converted back into pressure. This I know because the pressure in the stand-pipe is atmospheric pressure. This has to be because the top of the stand-pipe is the 3-inch vent pipe sitting on the roof of my house. The air sucked down the toilet bowl escapes through this 3-inch vent. If a bird's nest or snow clogs the vent, then the trapped air builds pressure in the stand-pipe. The back-pressure from the stand-pipe restricts the flow of water from the bowl, and the toilet can no longer flush properly.

This is an example of Bernoulli's equation in action. A steam vacuum ejector (jet) works in the same way. Centrifugal pumps and centrifugal compressors also work by converting velocity to pressure. Steam turbines convert the steam's pressure and enthalpy to velocity, and then the high velocity steam is converted into work, or electricity. The pressure drop we measure across a flow orifice plate is caused by the increase of the kinetic energy of the flowing fluid as it rushes (or accelerates) through the hole in the orifice plate.

Over the years I've purchased bigger and better homes. Now, Liz and I live in a house with seven bathrooms. Which is good, because at any given time, I almost always have at least one toilet mostly fully operational. Friends have asked why only two people need a house with seven bathrooms. Liz explains to them that, "If you ever tried to get my husband to fix anything, you would understand why Norm and I need a minimum of seven toilets in our home."

1.7 Distillation: The First Application

Extensive research has revealed that the best method to combat stress is alcohol. In 1980 I tried to become an alcoholic. Regrettably, I would fall asleep after my second drink. Ever since, I've had a desire to learn more about bourbon and scotch. In particular, in the production of a single malt scotch, how is the liquor separated from the barley mash?

Since 2003, I've been providing periodic process engineering services to a refinery in Lithuania. One evening after work, I was walking past the local village liquor store. Displayed in the window,

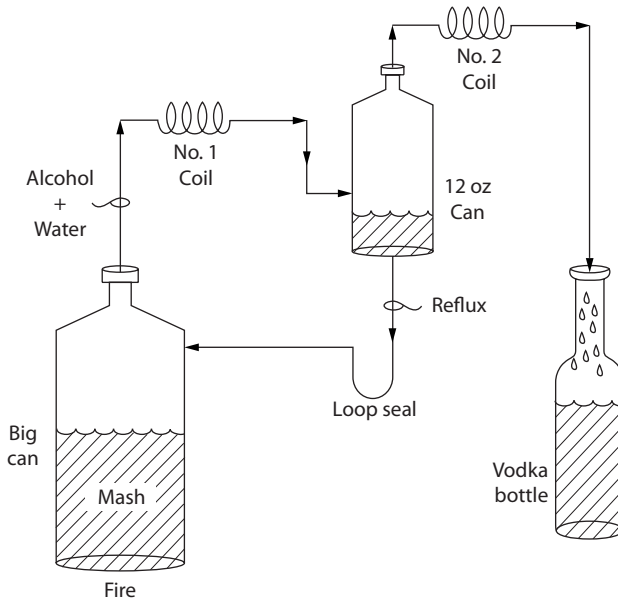


FIGURE 1.3 Vodka still: Lithuania, 2003. Device to separate alcohol from water.

surrounded by bottles of vodka, was a homemade still, as shown in Fig. 1.3. The two pots were just old soup cans. The big can containing the mash was about a gallon. The smaller can was 12 ounces. The appearance of the still suggested long use under adverse conditions. I'll provide a process description of this archaic apparatus.

The liquor in the big can is heated by a fire. The contents of the big can are:

- Water
- Alcohol
- Bad-tasting impurities

The objective is to produce vodka in the bottle of not less than 100 proof (that's 50 volume percent alcohol). Suppose that the bottle contains 80 proof (40 volume percent) alcohol. What can be done to bring the vodka up to the 50 percent spec?

There is only one thing that is under our control to change. This is the amount of firewood burned to supply heat to the big can. Should we add more heat to the big can or less heat?

If we add less heat to the big can, the vapor flow to the No. 1 condensing coil will diminish. As the water is less volatile than the alcohol, most of the reduction in vapor flow will be at the expense of water vaporization. Of course, there will be somewhat less vaporization of the more

volatile alcohol too. However, the primary effect will be to increase the percentage of alcohol in the vodka. This is good. The secondary effect is to reduce the vodka production. This is bad.

We would like to keep our product on spec (100 proof vodka) and also not lose production. To overcome this problem, we must first increase the heat input to the big can. To prevent the extra water vapor from diluting the vodka in the bottle, we must also increase the heat removed from the No. 1 coil. This is done by adding an additional length of coiled copper tubing to the No. 1 coil.

As a consequence of adding more heat to the big can, and also removing more heat from the No. 1 coil, more liquid will drain out of the 12-oz can, back to the big can. This liquid is called reflux. This reflux is revaporized in the big can and circulates back and forth between the big can and the 12-oz can. This recirculation helps to separate the lighter, more volatile alcohol from the heavier, less volatile water.

There are several ways to describe what is happening. As a chemical engineer, I would say that we are increasing the internal reflux ratio of the still. But what I would rather say is that we are making the still work harder. Harder in the sense that we are increasing both the reboiler heat duty and the condenser heat removal duty. By the reboiler duty, I mean to say the amount of firewood burning under the still. By the condenser heat removal duty, I mean the amount of heat radiating away to the air from the No. 1 condensing coil.

Why does making the still work harder decrease the water content of the vodka? Why does increasing the flow of reflux from the 12-oz can back to the big can improve separation efficiency between alcohol and water?

Well, if I reduced the heat to the still a lot, and removed the No. 1 coil (so that its heat removal duty was zero), then vapor would just blow through the 12-oz can. The water content of the vapor from the big can would be the same as the water content of the vodka in the bottle. The 12-oz can would then serve no purpose. However, as I partially condense the vapor flow into the 12-oz can, the water content of the vapors flowing into the bottle goes down, because water is less volatile than alcohol. The extra heat added to the big can prevents the extra heat removed by the No. 1 coil from reducing vodka production.

1.7.1 Two-Stage Distillation Column

The 12-oz can has a second function. Its main purpose is to separate the vapor flowing into the bottle from the water-rich liquid flowing back to the big can. The secondary function of the 12-oz can is to trap out bad-tasting impurities boiled out of the big can before they contaminate the vodka in the bottle.

The still pictured in Fig. 1.3 is acting as a two-stage distillation column—that is, a fractionator that has two fractionation trays.

The bottom tray is the big can. The top tray is the 12-oz can. If I wanted to build a similar facility in a chemical plant, I would have:

- Big can plus fire equals a once-through thermosyphon reboiler
- 12-oz can plus No. 1 coil equals a partial condenser and reflux drum
- Bottle plus No. 2 coil equals a final condenser and collector drum

Adding distillation trays between the reboiler and the partial condenser would permit the reflux to more efficiently wash back the bad-tasting impurities and water from the desirable and more volatile, up-flowing alcohol vapors.

1.7.2 The Loop Seal

Figure 1.3 shows a loop on the 12-oz can drain line. Without this loop, vapors flowing from the big can would partially bypass the No. 1 coil. Even worse, without the loop seal, the vapors flowing up through the reflux line would stop the flow of reflux. The 12-oz can would then fill up until it overflowed bad-tasting, watery vodka into our bottle.

This is like unsealing a tray's downcomer in a distillation tower. If the bottom edge of the downcomer from a tray is above the top edge of the outlet weir on the tray below, then vapor can blow up through the unsealed downcomer. This will prevent the internal reflux from draining down the column. Tower flooding and loss of product separation efficiency will result. This is called liquid flooding or excessive downcomer backup due to loss of the downcomer liquid seal.

1.7.3 Size of the Big Can

I explained that as we add more heat to the big can (more reboiler duty) and more capacity to the No. 1 coil (more condenser duty) that fractionation would get better. Better in the sense that the proof of the vodka would go up without reducing the production of vodka. Suppose, though, that the velocity of vapor leaving the big can becomes too great. We've all seen what happens when we boil soup too quickly in a small pot. The pot foams or floods over onto the stove. We should have used a bigger pot or we should have kept the heat low on the stove.

If the mash in the big can boils over or floods over, it will contaminate the 12-oz can with bad-tasting, water-rich liquid. If all this contaminated liquid cannot drain down through the loop seal fast enough, then the 12-oz can will also flood over into the bottle. The resulting weak, foul-tasting vodka will never be sold in Lithuania.

1.7.4 Jet Flood

The flooding over of the big can is rather similar to vapor or jet flood from a distillation tower tray. If the area of the tray is too small or if

the vertical separation (tray spacing) between the tray decks is inadequate for a particular vapor velocity, then the distillation tower will flood due to excessive entrainment of liquid from the tray below to the tray above.

If the liquid is full of particulates, like starch when I boil pasta, then the flooding will occur at a lower vapor velocity. When my pasta pot floods over, I turn down the heat. If my vodka still is flooding, then more heat will increase, and not decrease, the water content of my vodka bottle. By checking how the heat added to my still is affecting the proof of the vodka, I could determine the optimum amount of wood to burn under the still.

Same thing in a distillation column. If adding more reboiler duty and more reflux is making fractionation efficiency worse, then the column is suffering from flooding due to either downcomer backup or jet flood, or both. Note that the presence of particulates (dirt, rust, coke fines) reduces the capacity of distillation towers.

1.8 Origin of Reflux

As you read the subsequent chapters in this book, please recall the picture of the vodka still. Keep in mind that the origin of the reflux is the vapor generated from burning firewood beneath the big can. This concept is the key to understanding how distillation towers function. You cannot have more reflux without increasing your reboiler duty, unless you are willing to sacrifice the overhead product yield. And if extra reflux results in more heavy components in the overhead, the trays are flooding.

I have seen a working example of the first continuous distillation column in an apple orchard in southern England. It has a reboiler, feed preheater, reflux condenser, and four bubble cap distillation trays. The column produces apple brandy. We purchased a bottle of brandy from this orchard in 1996, but it remains unopened. Whenever I suggest that we open the bottle, Liz accuses me of trying to become an alcoholic.

1.9 Glossary

Many of the terms used in this text are defined in the following chapter. Otherwise, the first time we have used a new technical term, we have tried to show the term in *italics*, which is then defined at the back of the text in the Glossary. A lot of my problems as a young engineer were a result of my not being able to understand the vocabulary used in the process and refining industry. We have tried to address this problem in the Glossary and in Chap. 2.

CHAPTER 2

Basic Terms and Conditions

When I reported to work at Amoco's Whiting Refinery in 1965, the plant operators and my fellow engineers assumed that I understood a wide range of terms that actually had little meaning to me. Worse, I was suddenly confronted with the need to employ basic technical concepts that I thought I had mastered in high school. But when I tried to apply these concepts to solve plant problems, I realized that I had never really understood them. I've assembled a list of the concepts and terms that I had to quickly learn early in my career.

Work I was told in school that work is force times distance. I still have no idea what this means. To me, work is carrying bricks up a hill. If I carry 50 lb of bricks up a 100-ft hill, I have done 5000 ft-lb of work.

Power This is how fast I work. If I carry 50 lb of bricks up a 100-ft-high hill in an hour, then my power output is 5000 ft-lb per hour. If I do the job in 30 min, my power output has doubled.

Amperage Amps are a form of electric work. Motor amps are controlled by what the motor is driving, not the horsepower rating of the motor.

Kilowatts Watts are a form of electric power.

BTU This is the amount of heat needed to warm 1 lb of water by 1°F.

Latent heat This is the number of BTUs needed to change 1 lb of liquid into 1 lb of vapor. For water, this is about 1000 BTU per pound; for diesel oil, about 100 BTU per pound.

Sensible heat This is the heat associated with the temperature of the material. This does not include latent heat.

Specific heat This is the number of BTUs needed to heat 1 lb of liquid 1°F; for water, 1.0 BTU; for diesel oil, 0.5 BTU.

Friction A fluid moving through a pipe loses pressure. The lost pressure is converted to heat by friction inside the pipe. Friction converts work into heat. Converting heat back into work is much more difficult.

Enthalpy This is sensible heat plus latent heat.

Potential energy This is pounds of material elevated above a base level. Multiply the elevation (feet) times the pounds. The answer has the units of foot-pounds, just like work.

Kinetic energy This is the energy needed to accelerate a pound of material from a low velocity to a higher velocity.

Acceleration energy It takes more energy to make a fluid move faster than to keep it moving at a constant speed.

Momentum An important term: Momentum is mass times velocity. Alice weighs 100 lb and has the same momentum as Alan, who weighs 200 lb. If Alice is running at 8 miles per hour, how fast is Alan running? (Answer: 4 miles per hour.)

Energy Add up momentum, kinetic energy, potential energy, enthalpy, work, friction, and amps. The sum of all these is the total energy of the system. The total energy of the system remains constant when one sort of energy is transformed to another type.

Reaction heat I should have added that into the above list. If a chemical reaction such as burning wood liberates heat, it is called an exothermic reaction. Thermal cracking of hydrocarbons absorbs heat. That's an endothermic reaction.

Water hammer If I hammer a nail into a brick wall, the nail gets very hot. This is an example of converting the momentum of the hammer into sensible heat of the nail. When water flowing through a pipe is suddenly stopped, the resulting bang is called "water hammer." It's the conversion of the momentum of the water into pressure.

Mole of gas I have a box full of air. The air is at atmospheric pressure and 60°F. The size of the box is such that it contains 379 ft³. The box size was selected arbitrarily a long time ago. The number of molecules of air inside the box is called a "mole" (mol) of air. The number of molecules is about 10²³. A really big number.

Molecular weight If I weigh the air in this box, it will weigh 29 lb. Therefore the molecular weight of air is 29 lb/mol of air. If the box contained hydrogen, it would weigh 2 lb. Therefore the molecular weight of hydrogen is 2. If I displace the hydrogen with propane from an LPG (Liquefied Petroleum Gas) cylinder, the box weight would be 44 lb. The molecular weight of propane is 44 lb.

Gas laws As I double the absolute pressure of a gas, its volume is cut in half. As I double the absolute temperature of a gas, its volume doubles.

Absolute pressure This is gauge pressure plus 14.7 psi (if I am at sea level) or number of bars pressure plus 1.0. (Note: psi is Pounds per Square Inch.)

Absolute temperature This is °F temperature plus 460 R (Rankine) or °C temperature plus 273 K (Kelvin).

Compressibility Heavier gases are easier to compress than lighter gases. The compressibility factor for hydrogen is 1.0. For propane or butane, it's about 0.90.

Compression work Work needed to boost the pressure of a mole of gas. It takes less work to compress 44 lb (1 mol) of propane than it takes to compress 2 lb (1 mol) of hydrogen, partly because of their compressibility factors.

Compression ratio The compressor discharge pressure (in absolute pressure) divided by the suction pressure (also in absolute pressure). If I'm compressing air, the suction pressure is atmospheric pressure, and the discharge pressure is 29.4 psig, what is the compression ratio? (Answer: 3.)

Heat of compression Gases get hot when they are compressed. The hotter they get, the less efficiently the compressor is working. Bigger compression ratios also increase the temperature of the gas as it is compressed.

Expansion cooling You would suspect that when we depressure gas it would cool, if the gas gets hotter upon compression. Except for pure hydrogen, that is correct. Steam, air, and fuel gas all cool when we let the pressure down across a control valve.

Head A tank with 23 ft of liquid has a head of 23 ft regardless of the liquid density.

Head pressure A tank with 23 ft of water has a head pressure of 10 psi. A tank with 23 ft of gasoline has a head pressure of 7 psi, because gasoline is less dense than water. Its specific gravity is 0.70.

Specific gravity Water has a density of 62 lb/ft³. This density has been arbitrarily defined as a specific gravity—1.00 s.g. Since gasoline has a density that is 70 percent of water, its specific gravity is 0.70 s.g.

Temperature correction of specific gravity Hydrocarbon density drops by 5 percent for each increase of 100°F. For water it's 2 percent for each 100°F.

Viscosity Expressed in centipoises it is a fluid property measuring how much a fluid in a pipe drags along the walls.

Centipoises or centistokes Both terms have about the same value. A high viscosity fluid has 100 centipoises (cP) such as cold, heavy crude oil. A low viscosity fluid has 1 or 2 cP such as water or kerosene. Going from 2 to 20 cP would about double the pressure drop in a pipeline. Liquids over 100 cP or centistokes (cSt) are not easily pumped by a centrifugal pump. Increased temperature reduces viscosity.

Thermal conductivity This is the ability of a material to let heat pass. Metals, water, and materials that are good conductors of electricity have a high thermal conductivity. Air, rubber, and materials that are bad conductors of electricity have a low thermal conductivity. High viscosity hydrocarbons are bad conductors of heat.

Thermal expansion Railroad tracks grow longer in the heat of the sun. The hot tubes in an exchanger grow more than the cold shell. Hence, we have a

floating head in the tube bundle to accommodate differential rates of thermal expansion between the tube bundle and the shell.

Auto-ignition This is the temperature at which materials burn without a source of ignition. Gasoline auto-ignites at 450°F. Asphalt auto-ignites at 320°F, methane at 1000°F. The heavier the hydrocarbon, the lower the auto-ignition temperature. Iron sulfide (pyrophoric iron) burns at room temperature when it is dry.

Flash temperature Hold a flame over a cup of diesel fuel; it will start to burn at its 160°F flash temperature. Gasoline's flash temperature is below room temperature. Jet fuel's flash is 110°F. The lighter the hydrocarbon, the lower the flash temperature.

Vapor pressure This is a key concept. As we heat a liquid, the molecules in the liquid try to escape into the vapor phase. The hotter the liquid, the harder they try to escape. The pressure that the molecules of liquid create as they push out into the vapor space is the liquid vapor pressure. More volatile liquids such as LPG have a higher vapor pressure than less volatile diesel oil.

Boiling point Heat a liquid and its vapor pressure increases. When the liquid's vapor pressure equals the pressure in the vessel, the liquid starts to boil. The temperature at which this boiling starts is the liquid's boiling temperature.

Bubble point This is the same as boiling point. When a liquid is at its bubble point, it is also said to be a saturated liquid at the temperature and pressure. If we raise the pressure, the liquid's bubble point temperature also goes up.

Dew point A vapor at its dew point temperature is on the verge of starting to condense to a liquid. Cool the vapor by 1°F, or raise its pressure by 1 psi, and it will form drops of liquid. Air at 100 percent relative humidity is at its dew point temperature. Cool it by 1°F and it starts to rain.

Relative volatility Divide the vapor pressure of a lighter material by the vapor pressure of a heavier material. The bigger the resulting number, the larger the relative volatility. It's easier to separate two components in a distillation tower if they have a larger relative volatility.

Theoretical stage This stage is created when vapor and liquid mix perfectly and then separate without entraining any drops of liquid into the vapor or retaining any foam in the liquid.

Tray efficiency A tray in a distillation tower will work at some percentage of efficiency compared to a theoretical stage. This is mostly a function of how well the vapor and liquid are mixed on the tray deck, and how well they are then separated.

Gauge This is the thickness of metal. The smaller the gauge, the thicker the metal. Twelve gauge is about 0.1 inch thick.

Mesh This is the number of wires in a filter screen. The bigger the mesh, the finer the filter.

Flange I worked at Amoco Oil for nine years before I found out what is meant by a flange. It's the flat end of a pipe that is used to bolt up to a flange on another piece of piping. Bolts, with nuts at each end, are used to force the flanges together. A typical flange might have eight bolt holes.

Expansion loop Piping thermally expands as it gets hot. Allowance must be made for the growth in pipe length, otherwise the pipe will break by cracking at its welds. We burned down a fractionator at the Good Hope Refinery because of such an omission.

Spool piece Piping is made in sections with flanges at each end. This makes it easier to replace bad sections of pipe than if the piping sections were welded together, though welded pipe is cheaper.

Flange rating Connections on vessels, spool pieces, and valves have a pressure rating called a flange rating. This rating can be confusing—for example, a 150 psig flange rating is actually good for about 230 psig design.

Gasket This is the softer material that is pressed between flanges to keep them from leaking. Using the wrong gasket is a common cause of fires in process plants. Gaskets have different temperature and pressure ratings.

Pipe size Process piping comes in particular nominal sizes:

0.75 inch	—
1 inch	25 mm
2 inch	50 mm
2.5 inch	—
3 inch	80 mm
4 inch	100 mm
6 inch	150 mm
8 inch	200 mm
10 inch	250 mm

The nominal size does not refer either to the outside or the inside diameter of the pipe. Pipe thickness affects the internal diameter (ID).

Tube sizes Tubing sizes are entirely different from pipe sizes. Tubing is often used in heat exchangers and fired equipment like furnaces.

Screwed fittings These are used to assemble screwed connections and field instruments on pipes. There are:

- Pipe thread fittings
- Instrument or tubing fittings
- Metric fittings

None of these will screw together.

Relief valves These valves open to relieve excess pressure to protect a vessel from failure. Also called safeties or pop valves.

Flare system This is a piping network that runs through the plant to collect vents of gas so that they can be combusted at a safe location in the flare stack.

Utilities Most plants have some of the following utility systems connected to process units:

- Natural gas
- Nitrogen
- Plant air
- Instrument air
- Steam of various pressures
- Cooling water
- Service water
- Boiler feed water
- Fire water
- Fuel gas
- City water

Your company safety policy does not permit you to cross-connect these systems. Connecting natural gas to plant air killed 17 workers at a Louisiana refinery.

MAWP (maximum allowable working pressure) This is the legal maximum pressure that a process vessel is allowed to experience. Above this pressure, a relief valve should open to protect the vessel from catastrophic failure.

FLA (full-limit amps) This is the maximum amperage a motor should draw before it automatically trips off—rather like the 20-A circuit breaker at home.

Stress relief Coded vessels typically have a metal stamp attached that states, “Do not weld, stress relieved.” That means the vessel has been post-weld heat treated to remove stresses in the vessel wall created by welding during fabrication.

Alarms Process parameters (levels, temperatures, pressures, flows) are automatically controlled within a permissible range. If the parameter moves outside this range, it sometimes activates both an audible and a visual alarm. If the panel board operator fails to take corrective action, a trip may also then be activated.

Trip This is a safety device that automatically shuts down a piece of equipment. It’s a fail-safe mechanism often activated by unlatching a spring-operated valve, which then closes.

Gate valve This valve closes by sliding a plate or gate down between two grooves. Used to isolate different portions of the process equipment not used to control flow. The valve closes clockwise and takes about a dozen

turns to close. Ninety percent of the valves used in process plants are gate valves.

Plug valve This valve goes from 100 percent open to shut by turning a valve 90°. The natural gas supply to your house is shut off with a plug valve.

Control valve This valve is used to alter flows remotely. Normally it is moved by air pressure. A gate valve is sometimes used to control flows locally, but this wears out the valve and is best avoided.

Strobe light A modern tachometer used to measure rpm of rotating equipment. Old-style tachometers measured speed by measuring vibration frequency. They are called *reed tachometers*.

Dynamic machine Centrifugal pumps and centrifugal compressors are dynamic machines. They convert velocity imparted by the impeller to the fluid into fluid pressure.

Positive displacement machine A reciprocating compressor or gear type pump is an example of a positive displacement machine. It increases pressure by squeezing or pushing the fluid into a region of greater pressure.

Turbine A turbine uses steam pressure or burning gas to drive pumps and compressors at variable speeds. Motor drives are usually fixed-speed machines. Variable speed is an energy-efficient way to control flows by eliminating the downstream control valve.

Surge This is a terrifying sound that centrifugal compressors make when they malfunction either due to low flow or excessive discharge pressure, or low molecular weight gas.

Spillback A spillback allows fluid to recycle from the discharge back to the suction of a machine. It's one way to stop a centrifugal compressor from surging.

Cavitation When the pressure of liquid flowing into a centrifugal pump gets too low, liquid boils inside the pump case and generates bubbles. The discharge pressure and flow become erratically low.

Mechanical seal This is the part of a centrifugal pump that keeps the liquid from squirting out along the shaft. It is often subject to leakage due to pump vibration and cavitation.

AC motor Most of our pumps are driven by alternating current (AC), three-phase motors. Such motors that drive pumps are usually fixed-speed drivers. Direct current (DC) motors are rarely used in process plants.

NPSH (net positive suction head) The net positive suction head required to keep a centrifugal pump from cavitating. Cooling a liquid in a pump's suction line increases the pump's available NPSH, as does increasing the liquid level in the suction drum.

Entrainment Droplets of liquid are carried out of a vessel with the effluent vapors. High vapor velocities promote entrainment.

Flooding Massive entrainment of liquid caused by high-level or high velocity of the up-flowing vapor. A vapor-liquid separator (KO or Knock Out drum) that floods can wreck a downstream compressor in a few seconds.

Tapped-out This is a mechanism of flooding due to a high liquid level caused by flawed calibration of the level indicator. The liquid level has risen above the upper level indication connection point.

Level-trol This is the most common device used to measure levels in vessels. It works by measuring pressure differences between two level taps at different elevations.

Thermocouple A thermocouple consists of two wires of dissimilar metals. When joined and heated at their junction, a small electric current is generated. The hotter, the greater the current. That's how we measure temperature by measuring the current produced by a thermocouple. The thermocouple is housed inside a thermowell.

Orifice plate In an orifice plate, fluid flows through a metal plate with a hole. As it accelerates through this hole, the fluid pressure is reduced. The bigger the flow, the greater the ΔP . An orifice plate has a handle that sticks out of the piping showing the size of the orifice.

Pressure transducer A diaphragm is deformed inside a tube. This deformation generates an electric signal in proportion to the pressure inside the tube. The electric signal is used to monitor the pressure.

Draft A draft is the pressure difference inside a heater firebox and the surrounding air at the same elevation. The draft causes combustion air to flow through the burners and up the stack.

Radiant heat Inside a firebox, radiant heat is the heat that is transferred from the flames to the bricks and tubes by radiation from the flame. Most heat in a firebox is liberated as radiant heat. Sunlight is an example of radiant heat transfer.

Convective heat At the top of a firebox there is another section of the heater packed with tubes. Usually, these tubes have fins or studs. This is the section of the heater where convective heat transfer occurs. About 30 percent of heat recovered in a process heater is due to convective heat transfer. A sauna is an example of convective heat transfer.

Thermosyphon circulation (or natural circulation) Partially vaporizing a liquid causes it to flow to a higher elevation. The driving force is the density difference between the denser liquid feed and the less dense, partly vaporized effluent.

Forced circulation This is when a liquid is circulated through a heater or exchanger by means of a pump. A more complex and costly option as compared to thermosyphon circulation.

Gravity feed A kettle-type reboiler is an example of gravity feed. Flow is obtained by the liquid elevation above the exchanger. There is no

recirculation of liquid. Liquid simply flows from a higher elevation down to the exchanger where it is partially vaporized.

Reynolds number A measure of how turbulent a flow is. It's the linear velocity, times density, times internal pipe diameter, divided by viscosity.

Packed pump Centrifugal pumps that handle water or nonhazardous liquids do not use a mechanical seal. They use a cheaper and simpler packed seal.

Bernoulli's equation Conservation of energy. If we want higher fluid velocity, that extra energy must come from heat, pressure, or external work.

Bleeder A small drain ($\frac{3}{4}$ inch or $\frac{1}{2}$ inch) on a pump case, bottom of a vessel, or low point of a piping system. Must be kept capped or plugged.

MAWP Maximum allowable working pressure legally permitted for a vessel. The vessel will have an ASME code stamp. Heat exchangers are also rated with MAWP.

Thermowell The thermowell is a metal sleeve with a closed end inserted into a process vessel or process piping. A thermocouple is inserted inside the thermowell. Never unscrew the thermowell from a vessel that's in service.

Kettle reboiler Also called a *gravity feed reboiler*. It has only a vapor outlet nozzle venting back to the tower.

Bridle Same as level-trol or stand-pipe. Used to measure liquid levels in a process vessel.

Vortex breaker Located in draw-off nozzle sumps, or on the liquid outlet of the bottom of a vessel. Not required for low nozzle outlet velocities.

Nozzle Connections on the side of vessels and heat exchangers. Nozzles come in standard sizes corresponding to standard pipe diameters.

Vacuum jet nozzle Small precision-engineered steel nozzle located within a vacuum jet.

Entropy The measure of the potential ability of a fluid to do work. As the entropy of a fluid increases, its ability to do work decreases.

Grid tray Similar to a sieve tray. A perforated distillation tray deck, but with no movable valve caps. Represents modern distillation tray design.

Panel operator The operator assigned to control the unit. This individual will work 8- to 12-hour shifts and is normally regarded as the most important person on a process unit at any given time. Usually not an engineer, but an experienced plant operator.

Process engineer One who uses the principles of heat balance, hydraulics, vapor-liquid equilibrium, and chemistry to solve plant operating problems and optimize operating variables. Your authors are process engineers.

I hope that understanding these concepts and terms will help you in understanding the remainder of our book. Some concepts reappear in different forms. For example, check my descriptions of draft and thermosyphon circulation. They are really the same concept—heat that causes a density difference causes a flow. The application is different, but the concept is the same.

CHAPTER 3

How Trays Work: Flooding

History of Distillation

The ancient Egyptians produced beer from barley that had a few percent of alcohol. Next, wine was produced by fermenting grape juice, which had a greater sugar content than barley. This brought the content of alcohol up to about 14 percent. Next, fortified wines (like sherry or port) were made by adding extra sugar and yeast to the fermenting grape juice. This increased the alcohol up to about 17 percent. Much above this point, alcohol kills off the yeast.

Next, distillation was used. I visited a primitive distillation plant in Peru. It was a single-stage evaporation process. The fermented grape juice is partially vaporized and the alcohol–water vapors are totally condensed. The resulting condensate is 40 to 44 percent alcohol. The higher alcohol content is obtained by vaporizing less of the still’s content.

To go much beyond the 44 percent alcohol, one needs to introduce modern process engineering technology:

- Partial condensation
- Reflux
- Reboiler

I’ve shown a sketch of such a facility in Fig. 1.3. The idea is to generate reflux to improve the separation between water and ethanol. To generate reflux, a partial (rather than a total) condenser is required. Also, a way of adding more heat to the still, to match up with the capacity of the partial condenser, is needed. I’ve now introduced a complex control loop into my plant. Alcohol levels of 60-plus percent can be obtained with this two-stage evaporator.

Finally, we have the Patent Still, introduced by the Scots in the 1830s. Now, distillation trays equipped with bubble caps and feed preheat are used. I visited an apple orchard in England, which used

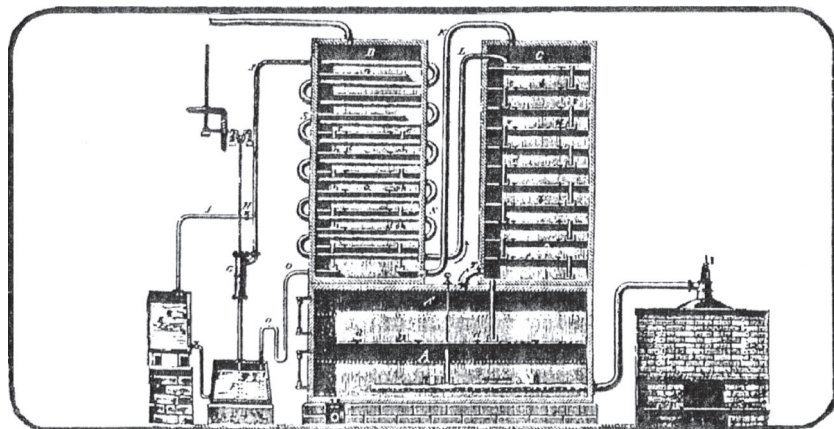


FIGURE 3.1 The very first distillation tower was the Patent Still. This drawing was filed with the original patent application submitted in 1835. Column on the right is a bubble-cap trayed tower. Column on the left is a feed preheater vs. an overhead vapor condenser.

the original design of the Patent Still to produce apple brandy. A sketch of this apparatus, whose design has not been altered in 180 years, is shown in Fig. 3.1. The sketch has been copied from the original patent application filed in London in the 1830s.

With the multitrayed distillation column, ethanol concentrations (as limited by the alcohol–water azeotrope) of 90-plus percent can be obtained, if enough reflux and enough trays are used.

3.1 Tray Types

Distillation towers are the heart of a process plant, and the working component of a distillation column is the tray. A tray consists of the following components, as shown in Fig. 3.2:

- Overflow, or outlet weir
- Downcomer
- Tray deck

There are two types of tray decks: perforated trays and bubble-cap trays. In this chapter, we describe only perforated trays, examples of which are

- Valves or flutter caps
- V grid, or extruded-valve caps
- Sieve decks
- Jet trays

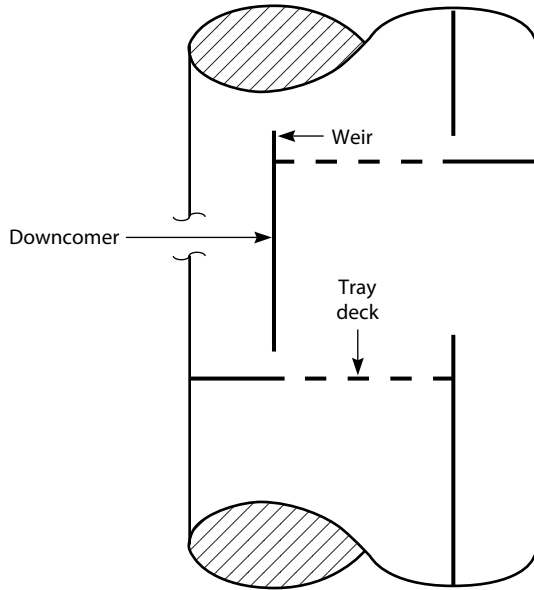


FIGURE 3.2 Perforated trays.

Possibly 90 percent of the trays seen in the plant are of these types. Perforated tray decks all have one feature in common; they depend on the flow of vapor through the tray deck perforations, to prevent liquid from leaking through the tray deck. As we will see later, if liquid bypasses the outlet weir and leaks through the tray deck onto the tray below, tray separation efficiency will suffer.

3.2 Tray Efficiency

Distillation trays in a fractionator operate between 10 and 90 percent efficiency. It is the process person's job to make trays operate as close to 90-percent efficiency as possible. Calculating tray efficiency is sometimes simple. Compare the vapor temperature leaving a tray to the liquid temperature leaving the trays. For example, the efficiency of the tray shown in Fig. 3.3 is 100 percent. The efficiency of the tray in Fig. 3.4 is 0 percent.

How about the 10 trays shown in Fig. 3.5? Calculate their average efficiency (the answer is 10 percent). As the vapor temperature rising from the top tray equals the liquid temperature draining from the bottom tray, the 10 trays are behaving as a single perfect tray with 100-percent efficiency. But as there are 10 trays, each tray, on average, acts like one-tenth of a perfect tray.

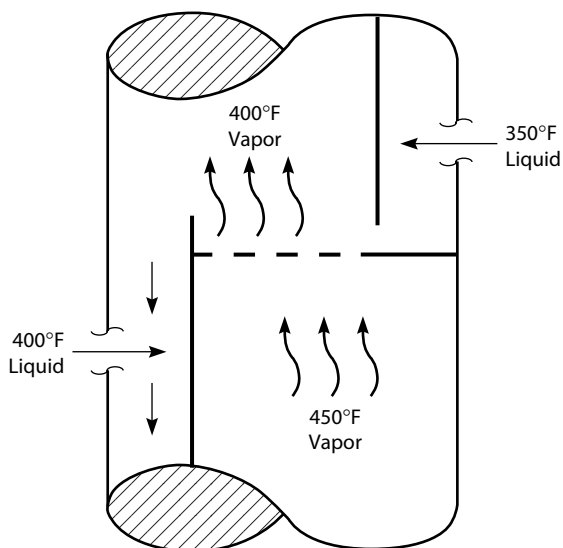


FIGURE 3.3 Hundred-percent tray efficiency.

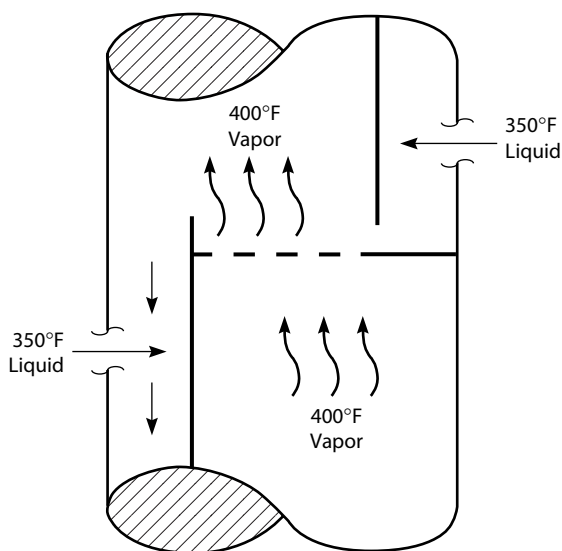
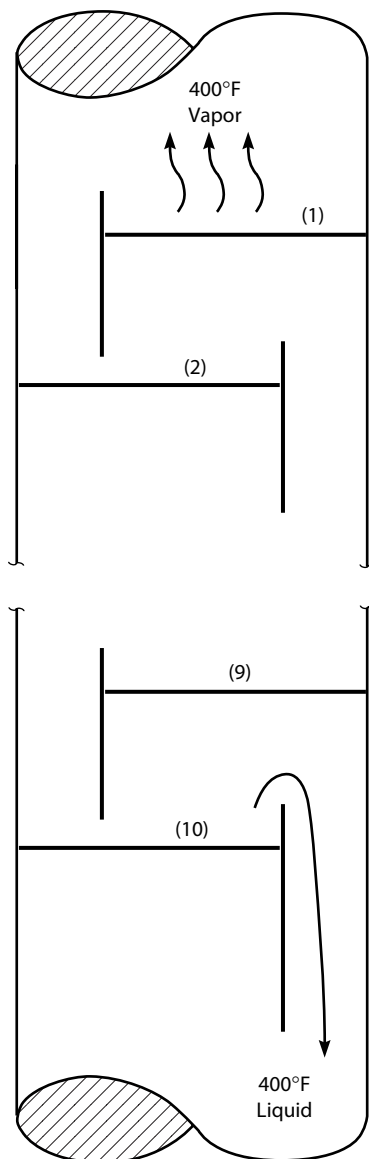


FIGURE 3.4 Zero-percent tray efficiency.

FIGURE 3.5
Average tray
efficiency =
10 percent.



Poor tray efficiency is caused by one of two factors:

- Flooding
- Dumping

In this chapter, we discuss problems that contribute to tray deck flooding.

3.3 Downcomer Backup

Liquid flows across a tray deck toward the outlet weir. The liquid overflows the weir, and drains through the downcomer to the tray below.

Vapor bubbles up through the sieve holes, or valve caps, on the tray deck, where the vapor comes into intimate contact with the liquid. More precisely, the fluid on the tray is a froth or foam—that is, a mixture of vapor and liquid. In this sense, the function of a tray is to mix the vapor and liquid together to form a foam. This foam should separate back into a vapor and a liquid in the downcomer. If the foam cannot drain quickly from a downcomer onto the tray below, then the foamy liquid or froth will back up onto the tray above. This is called *flooding*.

3.4 Downcomer Clearance

Referring to Fig. 3.6, note that the downcomer B is flooding. The cause is loss of the *downcomer seal*. The height of the outlet weir is below the bottom edge of the downcomer from the tray above. This permits vapor to flow up downcomer B. The up-flowing vapor displaces the downflowing liquid. That is, the vapor pushes the liquid up onto the tray above—which is a cause of flooding. On the other hand, Fig. 3.7 shows what happens if the bottom edge of the downcomer is too close to the tray below. The high pressure drop needed for the liquid to escape from downcomer B onto tray deck 1 causes the liquid level in downcomer B to back up onto tray deck 2. Tray 2 then floods. Once tray 2 floods, downcomer C (shown in Fig. 3.7) will also back up and flood. This process will continue until all the tray decks and downcomers above downcomer B are flooded.

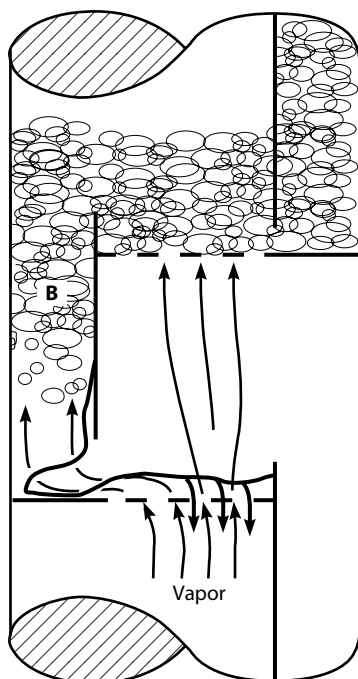
On the other hand, all trays in a tower below downcomer B will lose liquid levels and dry out when flooding starts in downcomer B. Thus, the following rules apply:

- When flooding starts on a tray, all the trays above that point will also flood, but trays below that point will go dry.
- An early indication of flooding in a distillation column is loss of liquid level in the bottom of the column.
- If the downcomer clearance—which means the distance between the bottom edge of the downcomer and the tray below—is too great, the downcomer becomes unsealed. Vapor flows up the downcomer, and the trays above flood.
- If the downcomer clearance is too small, then liquid backs up in the downcomer, and the trays above flood. To calculate the height of liquid in the downcomer, due to liquid flowing through the downcomer clearance:

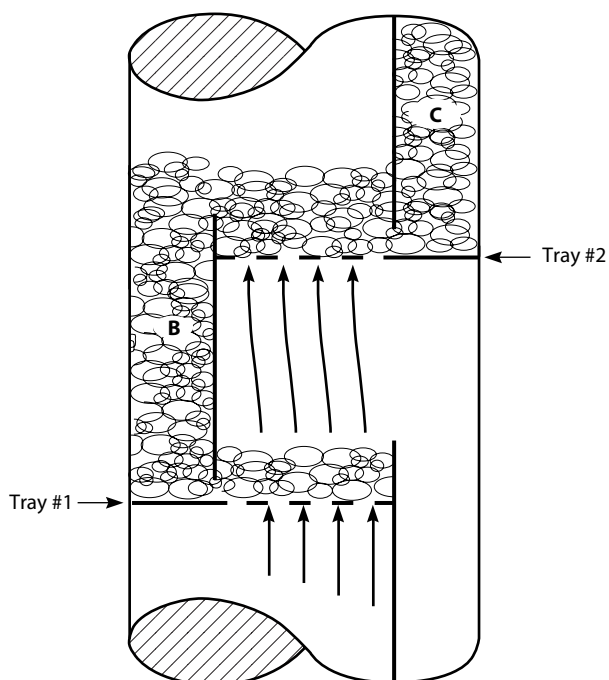
$$\Delta H = 0.6 \times V^2$$

FIGURE 3.6

Flooding due to lack of a downcomer seal.

**FIGURE 3.7**

Flooding caused by inadequate downcomer clearance.



where ΔH = inches of clear liquid backup in the downcomer, due to head loss under the downcomer

V = horizontal component of liquid velocity, in ft/s, as the liquid escapes from the downcomer

To guarantee a proper downcomer seal, the bottom edge of a downcomer should be about 0.5 inch below the top edge of the outlet weir. This dimension should be carefully checked by process personnel when a tower is opened for inspection. It is quite easy for sloppy tray installation to distort this critical factor.

3.4.1 Height of Liquid on Tray Deck

As the liquid level on a tray increases, the height of liquid in the downcomer feeding this tray will increase by the same amount. Again, excessive downcomer liquid or froth levels result in flooding and loss of tray efficiency.

The liquid level on a tray is a function of two factors:

- Weir height
- Crest height

The weir height on many trays is adjustable. We usually adjust the weir height to between 2 and 3 inches. This produces a reasonable depth of liquid on the tray to promote good vapor-liquid contact.

The crest height is similar to the height of water overflowing a dam. It is calculated from

$$\text{Crest height} = 0.4 (\text{GPM} \div \text{inch (outlet) weir length})^{0.67}$$

where crest height = inches of clear liquid overflowing the weir

GPM = gallons (U.S.) per minute of liquid leaving the tray

The sum of the crest height plus the weir height equals the depth of liquid on the tray deck. One might now ask, "Is not the liquid level on the inlet side of the tray higher than the liquid level near the outlet weir?" While the answer is "Yes, water does flow downhill," we design the tray to make this factor small enough to neglect.

3.5 Vapor-Flow Pressure Drop

We have yet to discuss the most important factor in determining the height of liquid in the downcomer. This is the pressure drop of the vapor flowing through the tray deck. Typically, 50 percent of the level in the downcomer is due to the flow of vapor through the trays.

When vapor flows through a tray deck, the vapor velocity increases as the vapor flows through the small openings provided by the valve caps, or sieve holes. The energy to increase the vapor velocity comes

from the pressure of the flowing vapor. A common example of this is the pressure drop we measure across an orifice plate. If we have a pipeline velocity of 2 ft/s and an orifice plate hole velocity of 40 ft/s, then the energy needed to accelerate the vapor as it flows through the orifice plate comes from the pressure drop of the vapor itself.

Let us assume that vapor flowing through a tray deck undergoes a pressure drop of 1 psi (lb/in²). Figure 3.8 shows that the pressure below tray deck 2 is 10 psig and the pressure above tray deck 2 is 9 psig. How can the liquid in downcomer B flow from an area of low pressure (9 psig) to an area of high pressure (10 psig)? The answer is gravity, or liquid head pressure.

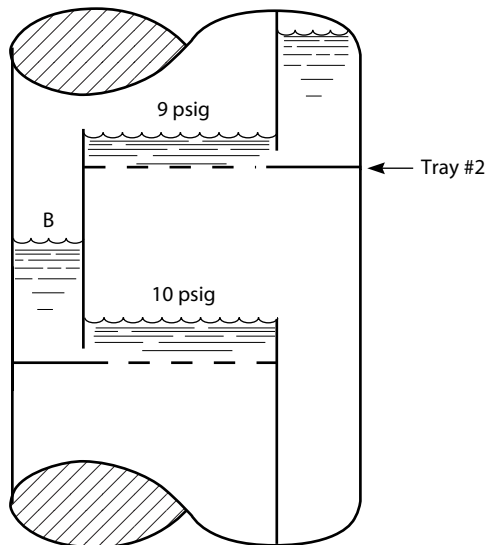
The height of water needed to exert a liquid head pressure of 1 psi is equal to 28 inches. of water. If we were working with gasoline, which has a specific gravity of 0.70, then the height of gasoline needed to exert a liquid head pressure of 1 psi would be 28 inches/0.70 = 40 inches of clear liquid.

3.5.1 Total Height of Liquid in the Downcomer

To summarize, the total height of clear liquid in the downcomer is the sum of four factors:

- Liquid escape velocity from the downcomer onto the tray below.
- Weir height.
- Crest height of liquid overflowing the outlet weir.
- The pressure drop of the vapor flowing through the tray above the downcomer. (Calculating this pressure drop is discussed in Chap. 4.)

FIGURE 3.8 Vapor ΔP causes downcomer backup.



Unfortunately, we do not have clear liquid, either in the downcomer, on the tray itself, or overflowing the weir. We actually have a froth or foam called *aerated liquid*. While the effect of this aeration on the specific gravity of the liquid is largely unknown and is a function of many complex factors (surface tension, dirt, tray design, etc.), an *aeration factor* of 50 percent is often used for many hydrocarbon services.

This means that if we calculated a clear liquid level of 12 inches in our downcomer, then we would actually have a foam level in the downcomer of $12 \text{ inches} / 0.50 = 24 \text{ inches}$ of foam.

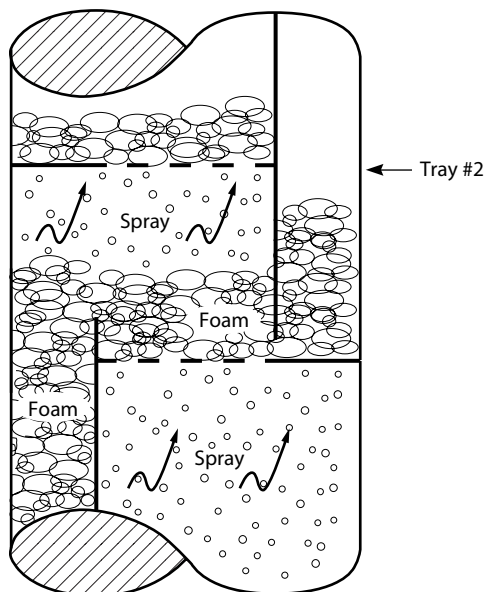
If the height of the downcomer plus the height of the weir were 24 inches, then a downcomer foam height of 24 inches would correspond to downcomer flooding. This is sometimes called *liquid flood*.

This discussion assumes that the cross-sectional area of the downcomer is adequate for reasonable vapor-liquid separation. If the downcomer loading (GPM/ft² of downcomer top area) is less than 150, this assumption is okay, at least for most clean services. For dirty, foamy services a downcomer loading of 100 GPM/ft² would be safer.

3.6 Jet Flood

Figure 3.9 is a realistic picture of what we would see if our towers were made of glass. In addition to the downcomers and tray decks containing froth or foam, there is a quantity of spray, or entrained liquid, lifted above the froth level on the tray deck. The force that

FIGURE 3.9
Entrainment
causes a jet flood.



generates this entrainment is the flow of vapor through the tower. The spray height of this entrained liquid is a function of two factors:

- The foam height on the tray
- The vapor velocity through the tray

High vapor velocities, combined with high foam levels, will cause the spray height to hit the underside of the tray above. This causes mixing of the liquid from a lower tray with the liquid on the upper tray. This backmixing of liquid reduces the separation, or tray efficiency, of a distillation tower.

When the vapor flow through a tray increases, the height of froth in the downcomer draining the tray will also increase. This does not affect the foam height on the tray deck until the downcomer fills with foam. Then a further increase in vapor flow causes a noticeable increase in the foam height of the tray deck, which then increases the spray height.

When the spray height from the tray below hits the tray above, this is called the *incipient flood point*, or the initiation of jet flooding. Note, though, that jet flood may be caused by excessive downcomer backup. It is simple to see in a glass column separating colored water from clear methanol how tray separation efficiency is reduced as soon as the spray height equals the tray spacing. And while this observation of the onset of incipient flood is straightforward in a transparent tower, how do we observe the incipient flooding point in a commercial distillation tower?

The reason I can write with confidence on this subject is that I worked with a 4-inch demonstration transparent column at the Chevron Refinery in Port Arthur, Texas, in 1989. I used the little distillation tower to explain to plant technicians how distillation towers worked. The tower's feed was windshield wiper fluid with blue dye. The alcohol went overhead and the blue water was the bottoms product.

3.7 Incipient Flood

3.7.1 A Fundamental Concept

Figure 3.10 illustrates the operation of a simple propane-butane splitter. The tower controls are such that both the pressure and bottoms temperature are held constant. This means that the percent of propane in the butane bottoms product is held constant. If the operator increases the top reflux flow, here is what will happen:

1. The tower-top temperature drops.
2. The amount of butane in the overhead propane product drops.
3. The tower-bottom temperature starts to fall.

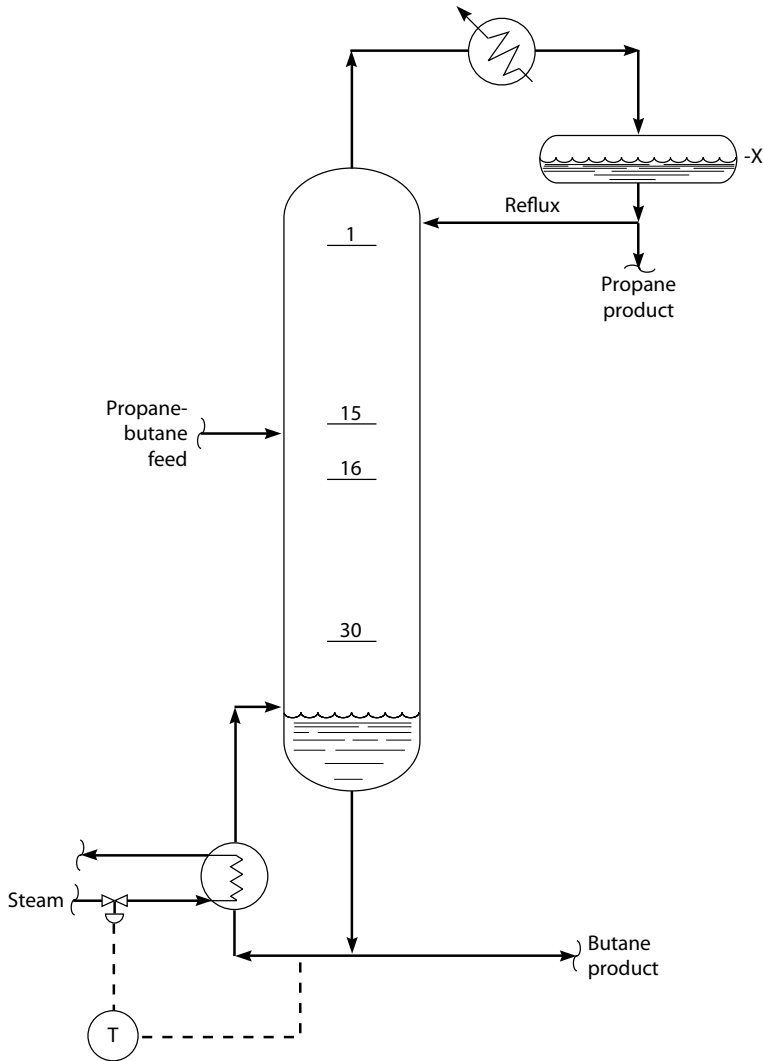


FIGURE 3.10 A simple depropanizer.

4. The reboiler duty increases, to restore the tower-bottom temperature to its set point.
5. The weight flow of vapor and the vapor velocity through the tray increase.
6. The spray height, or entrainment, between the trays increases.
7. When the spray height from the lower trays impacts the upper trays, the heavier, butane-rich liquid contaminates the lighter liquid on the upper trays with heavier butane.

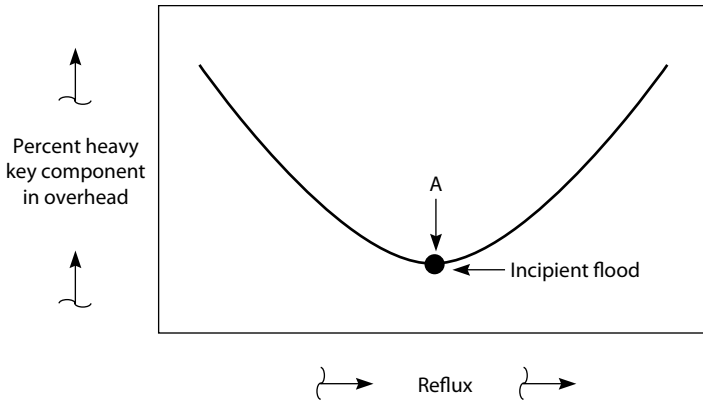


FIGURE 3.11 Definition of the incipient flood concept.

8. Further increases in the reflux rate then act to increase, rather than decrease, the butane content of the overhead propane product.

Figure 3.11 illustrates this point. Point A is called the *incipient flood point*, that point in the tower's operation at which either an increase or a decrease in the reflux rate results in a loss of separation efficiency. You might call this the *optimum reflux rate*; that would be an alternate description of the incipient flood point.

3.7.2 Bypassing Steam Trap Stops Flooding

I wake up early to answer email questions before breakfast. Here's today's question from South Africa:

- Hi Norman. We have a distillation tower that floods. Delta P on trays below feed point is stable; delta P above feed (trays 16–22), increase from 9 to 19 KPA. Condenser and reflux drum is internal in tower, and we cannot measure the reflux rate. Yesterday, bypassed steam trap on reboiler outlet, and flooding stopped. Conclude that flooding tower due to defective steam trap. What's your opinion? Note tower fractionation also improved after trap bypassed.

Regards,
Jon Sacha

- Dear Jon: You're quite wrong. When you bypassed the steam trap, you blew the condensate seal on the reboiler outlet. This permitted uncondensed steam to blow through the reboiler,

thus reducing the reboiler duty. The reduction in the reboiler duty reduced the vapor flow up the tower and hence the internal reflux rate. This unloaded the trays and stopped the flooding. Your observation that the tower fractionation improved as a consequence of bypassing the steam trap was a positive indication that you had degraded tray efficiency due to entrainment. That is, you were operating above the tower's incipient flood point. Certainly, there is nothing amiss with your reboiler steam trap. You should try to water wash the trays above the feed point onstream, as the trays in this service are typically subject to NH_4Cl salt sublimation. Hope this helps.

Best Regards,
Norman P. Lieberman

3.8 Tower Pressure Drop and Flooding

It is a characteristic of process equipment that the best operation is reached at neither a very high nor a very low loading. The intermediate equipment load that results in the most efficient operation is called the *best efficiency point*. For distillation trays, the incipient flood point corresponds to the best efficiency point. We have correlated this best efficiency point for valve and sieve trays as compared to the measured pressure drops in many distillation towers. We have derived the following formula:

$$\frac{(\Delta P)(28)}{(NT)(TS)(\text{s.g.})} = K$$

where ΔP = pressure drop across a tray section, psi

NT = the number of trays

TS = tray spacing, inches

s.g. = specific gravity of clear liquid, at flowing temperatures

On the basis of hundreds of field measurements, we have observed

$K = 0.18$ to 0.25 : Tray operation is close to its best efficiency point.

$K = 0.35$ to 0.40 : Tray is suffering from entrainment—increase in reflux rate noticeably reduces tray efficiency.

$K \geq 0.5$: Tray is in fully developed flood—opening a vent on the overhead vapor line will blow out liquid with the vapor.

$K = 0.10$ to 0.12 : Tray deck is suffering from low tray efficiency, due to tray deck leaking.

$K = 0.00$: The liquid level on the tray is zero, and quite likely the trays are lying on the bottom of the column.

$K = 1.00$: Tower is completely full of liquid.

3.8.1 Carbon Steel Trays

One of the most frequent causes of flooding is the use of carbon steel trays. Especially when the valve caps are also carbon steel, the valves have a tendency to stick in a partially closed position. This raises the pressure drop of the vapor flowing through the valves, which, in turn, pushes up the liquid level in the downcomer draining the tray. The liquid can then back up onto the tray deck and promote jet flood due to entrainment.

Of course, any factor (dirt, polymers, gums, salts) that causes a reduction in the open area of the tray deck will also promote jet flooding. Indeed, most trays flood below their calculated flood point, because of these sorts of problems. Trays, like people, rarely perform quite up to expectations.

The use of movable valve caps in any service where deposits can accumulate on the tray decks will cause the caps to stick to the tray deck. It's best to avoid this potential problem. Use of grid trays with fixed cap assemblies is preferred for most services.

3.9 Optimizing Feed Tray Location

From the design perspective, the optimum feed tray, for a feed with only two components, is that tray where the ratio of the two components matches the ratio in the feed. If the feed is at its bubble point temperature, then the feed temperature and the tray temperature will be the same, at the same pressure.

But that's only for a binary feed composition. In multicomponent distillation, the ratio of the key components in the feed will typically not coincide with the ratio of the key components in the liquid on the tray, even though the tray temperature is the same as the feed at its bubble point temperature.

So the question is, which of the following two criteria should be used to determine the feed tray location:

- Where the ratio of the key components in the liquid on a tray matches the ratio of the key components in the feed?
- Or where the feed temperature matches the temperature of the tray?

My practice is to identify both possible locations and then locate the feed nozzle halfway between the two options. I would provide two alternate feed nozzles at each of the other above locations, except the operators would likely never use them.

Incidentally, when I refer to "Key Components," I mean, for example:

- Debutanizer—Normal Butane and iso-Pentane
- Depropanizer—Propane and iso-Butane

- De-Ethanizer—Ethane and Propylene
- Gasoline Splitter—iso-Hexane and cyclo-Hexane

All of the above services have feeds with dozens of other non-key components.

3.10 Catacarb CO₂ Absorber Flooding

I've never told this story to anyone. Not even to Liz or my mom. It occurred in Lithuania in 2006. I had been hired to expand the capacity of the hydrogen plant that was limiting refinery capacity. The bottleneck was the absorber that removed CO₂ with catacarb solution from the hydrogen product. This absorber was subject to flooding as the catacarb circulation rate increased in proportion to H₂ production. That is, the solution was carried overhead with the hydrogen product.

I studied the design of the tower, but could not see an explanation for the flooding. Nevertheless, I decided to modify all the 40 trays in the absorber. The materials were ordered, and the labor force organized. However, the morning the absorber was opened, I received a call from my assistant, Joe.

"Hey, Norm, there's kind of a plate in front of the solution inlet nozzle (see Fig. 3.12). It don't show on the drawings. What you want to do with that plate?"

"Joe," I answered, "I'll be there in 10 minutes."

I looked at the plate. Dimension "x" was only about $\frac{3}{4}$ inch. Evidently, the plate was intended as an inlet solution distributor. I calculated that the delta P, as the solution flowed underneath the plate, was about 15 inches:

- Delta H (inches) = $0.6 (V)^2$
- Where V was the solution velocity through the $\frac{3}{4}$ -inch gap in feet per second.
- The plate was 12 inches high. As the solution rate increased, the liquid would back up over the top edge of the plate and be blown out of the top of the absorber. So, I told Joe to have the bottom 2 inches of the plate cut off to increase "x" to $2\frac{3}{4}$ inches.

"And, Joe," I continued, "Also, close up the tower afterward."

"Norm, but what about all the tower tray changes?" Joe protested.

"Don't argue. I know what I'm doing."

When the tower started up a week later, the Hydrogen Plant bottleneck was gone. The plant manager never found out what I did,

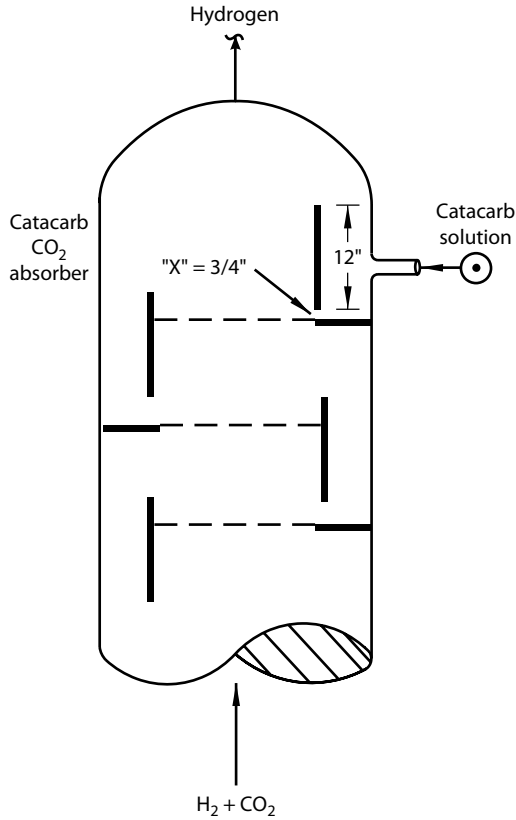


FIGURE 3.12 Restriction of the inlet distributor causes entrainment of the catacarb solution.

or that I had wasted \$20,000 for unused tray materials. Perhaps, since I had achieved the objective, he wouldn't have cared. Anyway, the alternate proposal to expand H₂ plant capacity, submitted by a major engineering contractor, would have cost \$3,000,000.

"All's well that ends well."

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CHAPTER 4

How Trays Work: Dumping

Weeping through Tray Decks

A distillation tray works efficiently when the vapor and liquid come into intimate contact on the tray deck. To this end, the liquid should flow evenly across the tray deck. The vapor should bubble up evenly through the perforations on the tray deck. The purpose of the outlet weir is to accomplish both these objectives, as follows:

1. Uneven liquid flow across the tray deck is particularly detrimental to good vapor-liquid mixing. For example, if half of the tray deck has stagnant liquid, then the vapor bubbling through the stagnant liquid cannot alter its composition.

Let me explain. A tray deck is a flat plate with holes. Liquid runs across the plate. Vapor bubbles up through the holes. If liquid only runs across part of this plate, vapor will still bubble up through the holes in the whole plate.

The vapor bubbling up through that portion of the tray deck where the liquid flow is active will mix with the flowing liquid. The flowing liquid will wash out the heavier components from the rising vapors.

On the other hand, the vapor bubbling up through that portion of the tray deck where the liquid flow is zero will also mix with the stagnant liquid. But it's like trying to wash dirty clothes in dirty water. The stagnant liquid cannot wash out the heavier components from the vapors, because it is already saturated with these heavier components.

Uneven liquid flow is promoted by the outlet weir being out of level. Liquid will tend to flow across that portion of the tray with a lower than average weir height. The portion of the tray upstream of the high part of the outlet weir will contain stagnant liquid. However, if the crest height (i.e., the height of

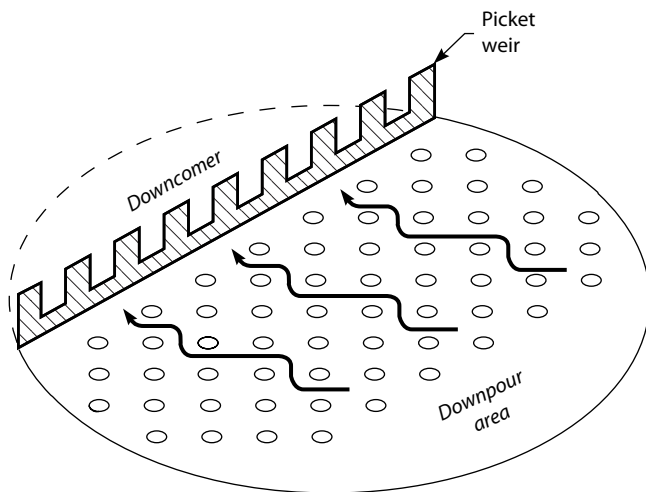


FIGURE 4.1 Picket weir promotes even liquid cross-flow at low flows.

liquid over the weir) is large, compared to the out-of-levelness of the tray, then an even liquid flow across the tray will result. To achieve a reasonable crest height above the outlet weir, a *weir loading* of at least 2 GPM per inch of weir length is needed. When liquid flows are small, the tray designer employs a picket weir, as shown in Fig. 4.1.

2. Uneven vapor flow bubbling up through the tray deck will promote vapor-liquid channeling. This sort of channeling accounts for many trays that fail to fractionate up to expectations. To understand the cause of this channeling, we will have to quantify total tray pressure drop.

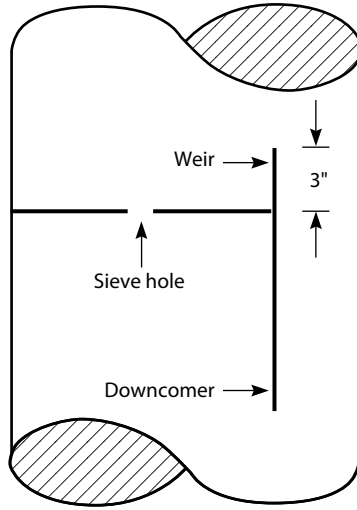
4.1 Tray Pressure Drop

4.1.1 Total Tray ΔP

Figure 4.2 shows a simple sieve tray with a single hole. Why is it that the liquid flows over the 3-inch outlet weir, rather than simply draining down through the sieve hole? It is the force of the vapor (or better, the velocity of the vapor) passing through the sieve hole that prevents the liquid from draining down the sieve hole. This is true whether we are dealing with a valve cap, extruded perforation, or a sieve hole. The valve cap does not act as a check valve to keep liquid on the tray.

FIGURE 4.2

A simplified sieve tray.



The author's industrial experience has proved this unpleasant fact on numerous occasions.

On the other hand, bubble caps (or even the more ancient tunnel cap trays) are different, in that they do not depend on the vapor flow to retain the liquid level on the tray deck. More on this later. For now, just recall that we are dealing only with perforated tray decks.

4.1.2 Dry-Tray Pressure Drop

For the force of the up-flowing vapor to stop the liquid from leaking through the sieve hole shown in Fig. 4.2, the pressure drop of the vapor flowing through the hole has to equal the weight of liquid on the tray deck. The weight of liquid trying to force its way through the sieve hole is proportional to the depth of liquid on the tray deck. The pressure drop of the vapor as it accelerates through the sieve hole is

$$\Delta P_{\text{dry}} = K \frac{D_V}{D_L} Vg^2$$

where ΔP_{dry} = dry tray pressure drop, in inches of clear liquid

D_V = density of vapor, lb/ft³

D_L = density of liquid, lb/ft³

Vg = velocity of vapor or gas flowing through the sieve hole, ft/s

K = an orifice coefficient, which can be as low as 0.3 for a smooth hole in a thick plate and 0.6 to 0.95 for various valve tray caps

4.1.3 Hydraulic Tray Pressure Drop

The weight of liquid on a tray is created by the weir height plus the crest height. We have defined the crest height (in inches of clear liquid) in Chap. 3, as

$$\text{Crest height} = 0.4 (\text{GPM} \div \text{inch outlet weir length})^{0.67}$$

The actual height of fluid overflowing the weir is quite a bit greater than we calculate with this formula. The reason is that the fluid overflowing the weir is not clear liquid, but aerated liquid—that is, foam. The fluid on the tray deck, below the top of the weir, is also foam. This reduces the effective weight of the liquid on the tray due to aeration. To summarize, the weight of liquid on the tray, called the *hydraulic tray pressure drop*, is

$$\Delta P_{\text{hyd}} = \text{AF} \times \text{WH} + 0.4 (\text{GPM} \div \text{inch outlet weir length})^{0.67}$$

where ΔP_{hyd} = hydraulic tray pressure drop, in inches of clear liquid

WH = weir height, inches

AF = aeration factor

GPM = gallons (U.S.) per minute

The aeration factor AF is the relative density of the foam to the density of the clear liquid. It is a combination of complex factors, but is typically 0.5.

4.1.4 Calculated Total Tray Pressure Drop

The sum of the dry tray pressure drop (ΔP_{dry}) plus the hydraulic tray pressure drop (ΔP_{hyd}) equals the total tray pressure drop (ΔP_{total}):

$$\Delta P_{\text{total}} = \Delta P_{\text{dry}} + \Delta P_{\text{hyd}}$$

expressed in inches of clear liquid.

When the dry tray pressure drop is significantly less than the hydraulic tray pressure drop, the tray will start to leak or weep and tray efficiency will be adversely affected.

When the dry tray pressure drop is significantly greater than the hydraulic tray pressure drop, the liquid on the tray can blow off of the tray deck and tray efficiency will be adversely affected.

For a tray to function reasonably close to its best efficiency point, the dry tray pressure drop must be roughly equal (± 50 percent) to the hydraulic tray pressure drop:

$$\Delta P_{\text{dry}} = \Delta P_{\text{hyd}}$$

This concept is the basis for tray design for perforated tray decks. While various valve tray vendors maintain that this rule does not hold for their equipment, it is the author's industrial experience that

valve trays leak just as badly as do sieve trays at low vapor hole velocities. To summarize:

$$\Delta P_{\text{total}} = K \frac{D_v}{D_L} V g^2 + AF \times WH + 0.4 (\text{GPM/inch outlet weir length})^{0.67}$$

4.2 Other Causes of Tray Inefficiency

4.2.1 Out-of-Level Trays

When trays weep, efficiency may not be significantly reduced. After all, the dripping liquid will still come into good contact with the up-flowing vapor. But this statement would be valid only if the tray decks were absolutely level. And in the real world, especially in large (>6-ft)-diameter columns, there is no such thing as a “level” tray. Figure 4.3 shows the edge view of a tray that is 2 inches out-of-level.

As illustrated, liquid accumulates on the low side of this tray. Vapor, taking the path of least resistance, preferentially bubbles up through the high side of the tray deck. To prevent liquid from leaking through the low side of the tray, the dry tray pressure drop must equal or exceed the sum of the weight of the aerated liquid retained on the tray by the weir *plus* the crest height of liquid over the weir *plus* the 2-inch out-of-levelness of the tray deck.

Once the weight of liquid on one portion—the lowest area—of a tray deck exceeds the dry tray pressure drop, the hydraulic balance of the entire tray is ruined. Vapor flow through the low area of the tray deck ceases. The aeration of the liquid retained by the weir on the low area of the tray deck stops, and hence the hydraulic tray pressure drop increases even more. As shown in Fig. 4.3, the liquid now drains largely

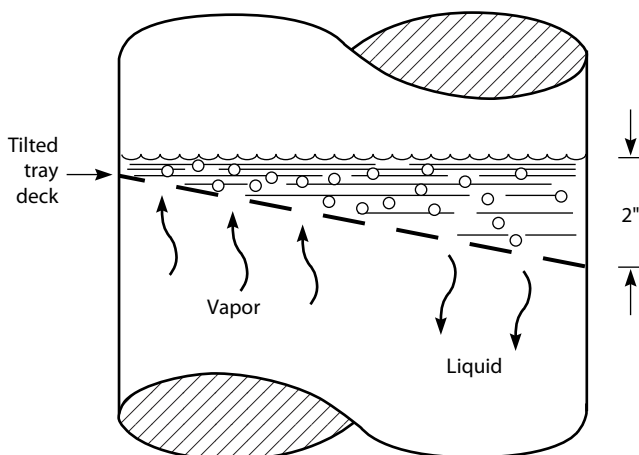


FIGURE 4.3 Out-of-level tray causing vapor-liquid channeling.

through the low area of the tray. The vapor flow bubbles mainly through the higher area of the tray deck. This phenomenon is termed vapor-liquid *channeling*. Channeling is the primary reason for reduced distillation tray efficiency, because the vapor and liquid no longer come into good, intimate contact.

The common reason for out-of-levelness of trays is sagging of the tray decks. Sags are caused by pressure surges and sloppy installation. Sometimes the tray support rings might not be installed level, or the tower itself might be out of plumb (meaning the tower itself may not be truly vertical).

4.2.2 Loss of Downcomer Seal

We stated in Chap. 3 that the top edge of the outlet weir is maintained about 0.5 inch above the bottom edge of the inlet downcomer to prevent vapor from flowing up the downcomer. This is called a 0.5-inch *positive downcomer seal*. But for this seal to be effective, the liquid must overflow the weir. If all the liquid is weeping through the tray deck, there will be no flow over the weir, and the height of the weir will become irrelevant. Figure 4.4 shows the result of severe tray deck leakage:

1. The downcomer seal is lost on tray deck 1.
2. Vapor flows up the downcomer between tray decks 1 and 2.

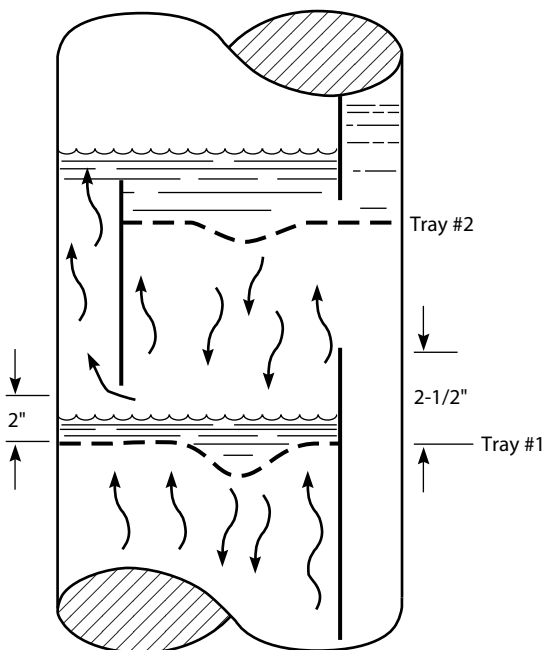


FIGURE 4.4 Sagging tray ruins downcomer seal.

3. Liquid flow is backed up onto the tray above, i.e., onto tray deck 2.
4. The dry tray pressure drop through tray 2 decreases due to low vapor flow through the tray deck.
5. The hydraulic tray pressure drop on tray 2 increases due to increased liquid level.
6. Tray 2 will now start to weep, with the weeping concentrated on the low area of the tray.
7. Tray 2 now has most of its vapor feed flowing up through its outlet downcomer, rather than the tray deck, and most of its liquid flow is leaking through its tray deck.

The net result of this unpleasant scenario is loss of both vapor-liquid contacting and tray efficiency. Note how the mechanical problems (i.e., levelness) of tray 1 ruins the tray efficiency of both trays 1 and 2.

4.2.3 Flooding at Reduced Vapor Rates

Referring to Fig. 4.4, if a tray deck is leaking so badly that none of the liquid is overflowing its weir, then the downcomer from the tray above will become unsealed. Then, due to vapor displacing the liquid in this downcomer, all the trays above the unsealed downcomer will flood. This gives rise to an unusual and rather confusing situation. That is, lower vapor rates, which cause a reduced ΔP through the trays, can cause flooding. Worse yet, the way one stops this flooding is to increase the ΔP across the sagging tray deck by increasing the vapor flow, for instance, by increasing the reboiler duty or stripping rate.

This contradicts common sense that flooding is promoted by higher vapor rates. But I have stopped flooding in a crude distillation tower recently in a Kansas refinery by increasing the stripping steam rate. Also, I have often seen that increasing vapor rates in a column reduces tray ΔP . I assume this has happened because I have accidentally resealed the downcomer above the out-of-level tray deck.

4.3 Bubble-Cap Trays

The first continuous distillation tower built was the "Patent Still" used in Britain to produce Scotch whiskey, in 1835. The patent still is to this day employed to make apple brandy in southern England. The original still, and the one I saw in England in 1992, had ordinary bubble-cap trays (except downpipes instead of downcomers were used). The major advantage of a bubble-cap tray is that the tray deck is leakproof. As shown in Fig. 4.5, the riser inside the cap is above the top of the outlet weir. This creates a mechanical seal on the tray deck, which prevents liquid weeping, regardless of the vapor flow.

Bubble-cap trays may be operated over a far wider range of vapor flows, without loss of tray efficiency. It is the author's experience that

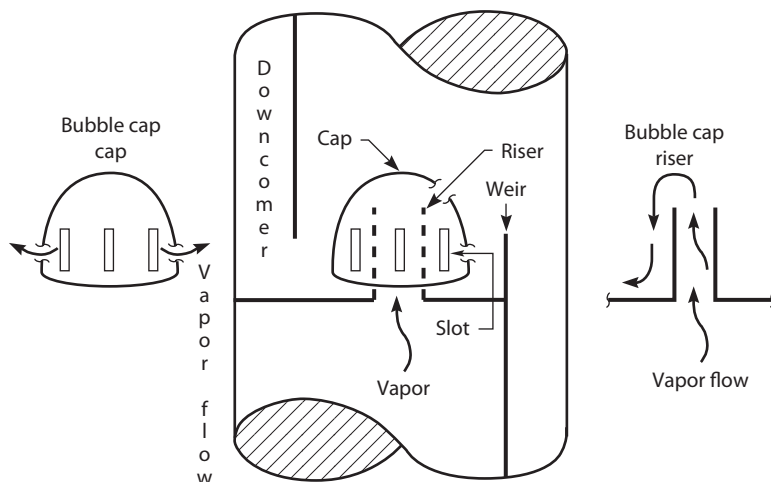


FIGURE 4.5 A bubble cap showing vapor pathway in operation.

bubble-cap trays fractionate better in commercial service than do perforated (valve or sieve) trays. Why, then, are bubble-cap trays rarely used in a modern distillation tower?

There really is no proper answer to this question. It is quite likely that the archaic, massively thick, bolted-up, cast-iron bubble-cap or tunnel-cap tray was the best tray ever built. However, compared to a modern valve tray, bubble-cap trays

- Were difficult to install, because of their weight.
- Have about 15 percent less capacity because when vapor escapes from the slots on the bubble cap, it is moving in a horizontal direction. The vapor flow must turn 90°. This change of direction promotes entrainment and, hence, jet flooding.
- Are more expensive to purchase.
- Are more difficult to clean.

But in the natural gas fields, where modern design techniques have been slow to penetrate, bubble-cap trays are still widely employed to dehydrate and sweeten natural gas in remote locations.

4.3.1 Distillation Tower Turndown

The problem we have been discussing—loss of tray efficiency due to low vapor velocity—is commonly called *turndown*. It is the opposite of flooding, which is indicated by loss of tray efficiency at high vapor velocity. To discriminate between flooding and weeping trays, we

measure the tower pressure drop. If the pressure drop per tray, expressed in inches of liquid, is more than three times the weir height, then the poor fractionation is due to flooding. If the pressure drop per tray is less than the height of the weir, then poor fractionation is due to weeping or dumping.

One way to stop trays from leaking or weeping is to increase the reflux rate. Assuming that the reboiler is on automatic temperature control, increasing the reflux flow must result in increased reboiler duty. This will increase the vapor flow through the trays and the dry tray pressure drop. The higher dry tray pressure drop may then stop tray deck leakage. The net effect is that the higher reflux rate restores the tray efficiency.

However, the largest operating cost for many process units is the energy supplied to the reboilers. We should therefore avoid high reflux rates, and try to achieve the best efficiency point for distillation tower trays at a minimum vapor flow. This is best done by designing and installing the tray decks and outlet weirs as level as possible. Damaged tray decks should not be reused unless they can be restored to their proper state of levelness, which is difficult, if not impossible.

4.4 New High Capacity Trays

All vendors now market a high capacity tray. These trays have a 5 to 15 percent capacity advantage over conventional trays. Basically, the idea behind these high capacity trays is the same. The area underneath the downcomer is converted to bubble area. This increase in area devoted to vapor flow reduces the percent of jet flood.

But what keeps vapor from blowing up the downcomer? What prevents loss of the downcomer seal? If the downcomer seal is lost, surely the downcomer will back up and flood the upper trays of the column.

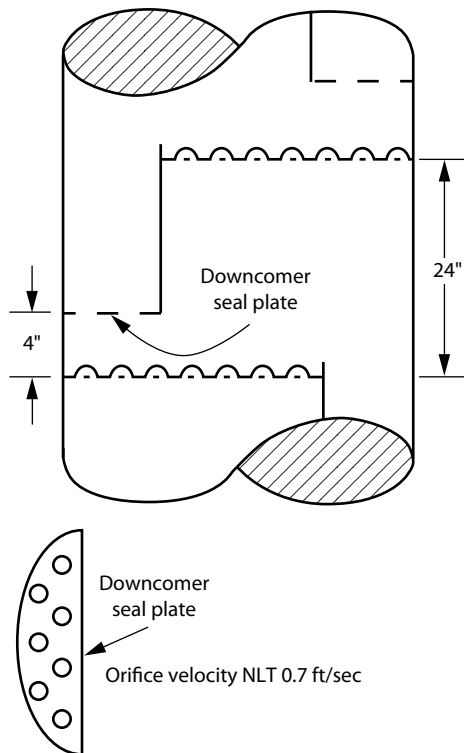
The design I'm most familiar with is the NorPro high capacity tray shown in Fig. 4.6. The head loss through the orifice holes in the downcomer seal plate shown is sufficiently high to prevent loss of the downcomer seal. These trays flood rather easily when their design downcomer liquid rates are exceeded. However, when operated at design downcomer liquid rates they perform very well indeed, and have shown quite a high vapor-handling capacity as compared to conventional trays.

The downcomer seal plate shown in Fig. 4.6 is an example of a dynamic downcomer seal. The Koch-Glitsch "Nye" tray also uses a dynamic downcomer seal to increase vapor-handling capacity. All trays with a dynamic downcomer seal suffer from two disadvantages:

- Loss of flexibility in that the liquid rates cannot be varied over too great a range without either flooding or unsealing the downcomers.

FIGURE 4.6

High capacity tray with downcomer seal plate.



- Tray installation complexity is always increased, sometimes with terrible consequences.

For these reasons, high capacity trays using dynamic downcomer seals are best avoided on new columns. They should be reserved for use on retrofit tower expansion projects.

4.5 Calculating Tray Efficiency

The reader with a chemical engineering background will observe that I've neglected the calculation of tray efficiency. There are three methods available to calculate tray efficiency:

- Overall tray efficiency
- Murphy tray efficiency
- Mass transfer tray efficiency

I've never had occasion to calculate tray efficiency for any distillation tower. I'm guided by observations on similar existing services. Certainly, tray hydraulics as described above (i.e., levelness, weeping,

entrainment) is the dominating factor in determining tray efficiency. However, when I run a computer simulation for a distillation column, and must then select a tray efficiency, I'm guided by the following:

- Absorption of ethane, propane—50%
- De-ethanizer, demethanizer—60%
- Sour water stripper for H_2S , NH_3 —50%
- Debutanizer—80%
- Crude distillation—90%
- Amine regeneration—70%
- FCU, Coker fractionators—90%
- Pumparounds—80%
- Vacuum distillation hydrocarbons—70%
- Steam stripping hydrocarbons—50%

I don't have a solid basis for any of the above parameters. The values quoted are mostly what I've seen other designers use or have experienced myself. Large negative variations from these parameters are rarely due to any calculation problems. Almost always, the problem is bad tray design or sloppy installation, or both.

The majority of the towers I work with have too many trays. Designing for extra trays is the designer's futile attempt to offset tray hydraulic problems described in this chapter.

Tray inspection after installation or during a unit turnaround is a lot more important than the selection of the best tray efficiency calculation model.

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CHAPTER 5

Notes on Tray Design Details

Most of my clients are careless regarding tray design. For example, their engineering department will run a tower computer simulation. Size the tower accordingly at 85 percent of jet flood and email the results to the tray vendors. The tray vendors cannot be trusted to work through the details that ensure a good tray design. Thus, after you have done your computer simulation, you need to write up a set of tray specs for the tray bid package with the appropriate details.

I have described these required details in the following notes, as I've taught them in my seminar, which, if you are reading this, you have likely already attended, but may already have forgotten?

5.1 Process Design Equipment Details

5.1.1 Tray Deck Design

For trays to work efficiently, I use the following criteria:

1. Weir height = $2\frac{1}{2}$ to 3 inches.
2. Hole or perforated area = 4 to 14 percent of bubble area. By bubble area, I'm referring to the tray area excluding the area under the downcomer.
3. To select the hole area, I'll first calculate the weight of liquid on the tray. This consists of two parts: the weir height and the crest height.
 - a. Since there is foam, not liquid on the trays, I assume the foam is half as dense as clear or flat liquid. So I divide the weir height by two.

- b. Then I calculate the crest height, or the height of the liquid (inches) over the top edge of the weir:

$$0.4 (\text{GPM} \div \text{inches of weir length})^{0.67}$$

- c. The weight of liquid on the tray is step *a* plus step *b*, above.
4. For perforated tray decks (sieve or grid trays), I calculate the pressure drop of the vapor flowing through the holes (inches). The pressure drop I want is the weight of liquid on the trays that I calculated above in step *c*. The idea is to keep the tray from leaking through the tray deck perforations.
5. The vapor ΔP (inches) is:

$$(K) \cdot (\text{Vapor Density} \div \text{Liquid Density}) \cdot (\text{Vapor Velocity})^2$$

where vapor velocity is for the vapor flowing through the holes in feet per second.

6. *K* for sieve trays is 0.3; *K* for grid trays is 0.6. Only an idiot would use movable valve trays. The valve caps stick to the deck and do not greatly retard tray deck leakage.

To summarize, I'll select a hole area for the tray, so that the velocity of vapor flowing through the holes will be big enough to keep the tray from leaking. Of course, if the tray decks are badly out of level, the above calculations are meaningless. So don't forget to inspect your tray installation for tray deck levelness (see my book, *Process Equipment Malfunctions*, McGraw-Hill, 2011).

5.1.2 Weir Design

Weir loading in U.S. gallons per minute (GPM) per inch of weir length should be kept between 3 and 12. Above 13 or 14, a loss of tray efficiency due to entrainment (i.e., jet flood) can be expected. Below 3 GPM per inch of weir, the liquid flow across the tray will not be particularly uniform and a loss in tray efficiency will result.

1. If the weir loading is excessive, increase the number of tray deck passes from one to two, or from two to four. I don't use three-pass trays, because they lack symmetry.
2. Increasing the number of passes reduces the flow path length between the liquid flowing from the inlet side of the tray to the weir. This reduces tray efficiency for flow paths of less than around 18 inches. So, to avoid this problem, I'll use a swept-back downcomer (Fig. 5.1) to decrease my weir loadings when my flow path length is getting too small. The tray vendors will have to work out the details of the swept-back downcomer.
3. If the weir loading is too small, picket weirs are needed to increase the loading to 4 GPM/inch. A picket weir is a strip of

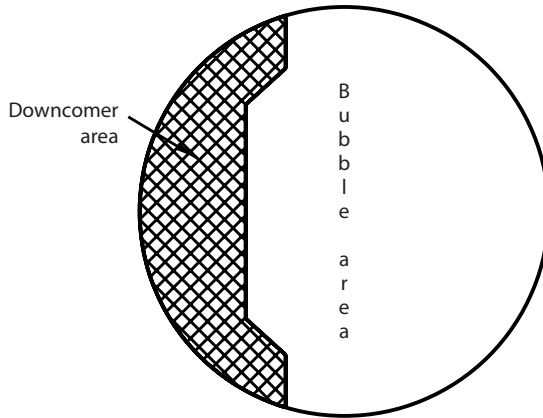


FIGURE 5.1 Top view of a swept-back downcomer.

steel welded to the top edge of the existing weir; each picket is 2 to 8 inches wide and around 12 inches high (Fig. 5.2). But in practice all the pickets are usually made by cutting rectangular notches at regular intervals into a single strip of metal, which then spans the full width of the weir and is attached to the existing weir. Process-wise, the end result is the same and still looks like Fig. 5.2.

I can't overemphasize how important picket weirs are to avoid weir loadings of less than 1 or 2 GPM/inch. Of course, if the weirs are not level, then the above criteria are meaningless. So inspect your trays for weir levelness. However excellent the design, if trays, weirs, or downcomers are not properly installed, then the design is irrelevant.

Picket weirs also even out the depth of liquid on the tray itself, and thus promote more uniform flow of vapor through the tray, which also enhances tray efficiency.

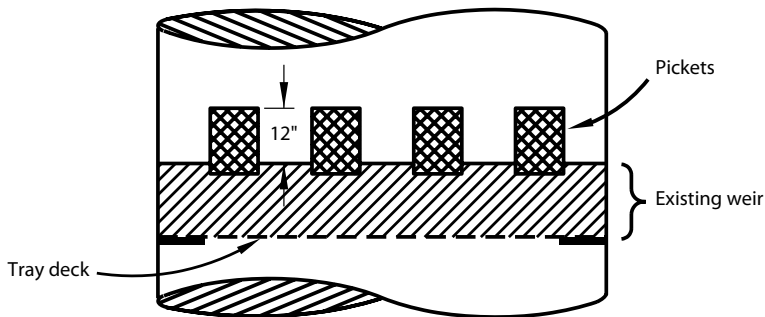


FIGURE 5.2 A picket weir.

5.1.3 Tray Ring Area

The tray support rings are typically 3 inches wide. That means there are no holes in the tray decks about 4 inches from the vessel wall. Liquid can bypass the bubbling vapor through these gaps and reduce tray efficiency. I have two ways of dealing with these gaps:

1. Drill ½-inch holes through the tray and tray ring. This will also marginally increase tower capacity.
2. Place a 3-inch-high by 4-inch-wide dam next to the vessel wall (i.e., four dams are needed for a two-pass tray).

For large-diameter trays, a similar gap will result where the tray panels are attached to the cross I-beam supports. Holes and/or dams are appropriate for these bypass areas as well.

Drilling holes through the edge of the tray and through the tray ring is an especially nice way to coax a bit more capacity out of small-diameter towers.

5.1.4 Increasing Tower Bubble Area to Reduce Jet Flood

Here's a neat trick that I learned from Dale Nutter, whose dad founded Nutter Engineering, the forerunner of Sulzer in the United States. It's worth perhaps 5 percent (max) of tower percent of jet flood capacity.

As shown in Fig. 5.3, create a ¼-inch gap between the downpour plate and the inlet perforated tray panel.

Vapor from the tray below escapes through the gap, passes toward the bubble area, and thus somewhat uses the tower area below the solid downpour plate. It's kind of like a poor man's "Nye"

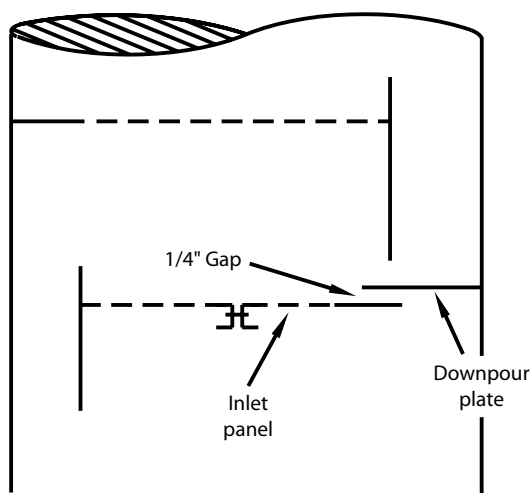


FIGURE 5.3 Quarter-inch gap increases tower capacity a few percent.

panel. Dale suggested using ¼-inch-thick wedges. I've just used ¼-inch steel washers.

5.1.5 Changing to Thicker Trays

I like to use thicker trays than other designers, mainly for mechanical integrity. As one increases the tray deck thickness, the pressure drop of the vapor flowing through the tray perforations goes down, not up. That's because the K value (i.e., orifice coefficient) is reduced. The effect on the tray performance is to promote tray deck leakage, but also marginally to increase tray capacity. Going from 2-mm tray thickness (about 14 gauge) to ¼-inch-thick trays will cut the vapor ΔP through the orifice by a lot. So be careful.

5.1.6 Tray Spacing

Tray capacity will increase with the square root of tray spacing. I use a 24-inch tray spacing for smaller diameter towers (3–8 ft), 30-inch tray spacing for midsize towers (9–12 ft), and 36-inch tray spacing for anything larger. Bigger spacing is needed to accommodate the cross I-beam supports.

One of my favorite tricks on H_2 hydro-desulfurizer recycle gas H_2S absorbers or sour water strippers, or any tower that has far too many trays, is to double the tray spacing. This permits me to increase tower capacity by 40 to 50 percent. Recycle gas H_2S absorbers, especially, have typically four times the number of trays required (see my book, *Process Engineering for a Small Planet*, Wiley, 2010).

Do not use tray spacing of less than 24 inches. These trays are hard to inspect. It's too difficult to crawl between the tray decks.

5.1.7 Downcomers

For 24-inch tray spacing, a maximum downcomer loading is 175 GPM per square foot of the downcomer top area. That's for clean services. For foaming services, I use 90 GPM. It does not hurt anything to make the downcomer too big. That is, there is no minimum downcomer loading.

It is industry-accepted practice to slope the downcomer, so that the bottom edge of the downcomer is only 60 percent of the top area. This increases the bubble area by maybe 5 to 7 percent. I never do this on new towers. It's an unnecessary risk and installation complication. However, for retrofits, it's not a bad idea.

One thing I never do in fouling services is to use recessed sumps below the downcomers. What's to prevent these sumps from filling up with sludge?

5.1.8 Downcomer Width

Having a downcomer with adequate area may not be sufficient to prevent downcomer flooding. If the downcomer is long and narrow,

it may still flood at low liquid rates. The problem is that the liquid cascading over the weir will impact the wall of the vessel. If the weir loading is high (perhaps 10 GPM/inch) and the downcomer width is narrow (perhaps 6 inches), this problem may occur. I'm not too sure of the parameters. However, I once increased the downcomer width on a tower but without increasing the downcomer area and stopped the flooding. The service was a fuel gas H_2S amine absorber, which is typically a nasty foaming service.

5.1.9 Foaming Services

If you think that your tray will be in a foamy service, it's best to double the usual downcomer area from 175 GPM per ft^2 to 90 GPM per ft^2 . Towers working as demethanizers or de-ethanizers fall into this category. If I have a sour water stripper with hydrocarbon liquids in the stripper feed or water in a hydrocarbon fractionator, I would anticipate foaming. The foaming service I know most about is H_2S absorption in amine. The greater the concentration of the amine, the greater the foaming. When I noted the 90 GPM per ft^2 value above, I had in mind 25 to 30 percent DEA or 20 percent MEA.

Refrigerated absorbers are also, in my experience, somewhat foamy, as is any system that is contaminated with particulates. Sand in recovered heavy bitumen is one such example.

5.1.10 Rounded Downcomer Outlets

This is a retrofit trick and not appropriate for new towers. The bottom edge of a downcomer is rounded rather than straight, as shown in Fig. 5.4.

The head loss under the downcomer (inches of liquid) is calculated as:

$$\text{Head Loss} = K \cdot V_1^2$$

where V_1 = feet/sec, the horizontal component of velocity underneath the downcomer.

K is reduced from 0.6 to 0.3 by using the rounded downcomer outlet.

To obtain this feature from the tray vendor, you need to specify a "Full Radius Downcomer Outlet." This will lower the liquid level in the downcomer by an inch or so. If you're limited by downcomer backup, this would provide about an extra 5 percent of capacity. No great improvement. But it doesn't cost very much either.

Of course, anything that reduces the head loss (i.e., delta P) of liquid flow past the bottom edge of the downcomer increases the possibility of unsealing the downcomer, as I'll describe in the following sections.

5.1.11 Setting the Downcomer Clearance

Specifying the downcomer clearance above the tray below is one of our more critical design criteria. If the clearance is too small, the

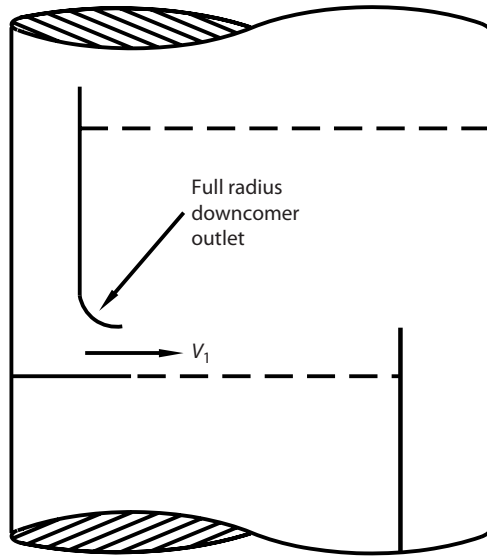


FIGURE 5.4 Rounded downcomer outlet.

downcomer will back up and flood. If the clearance is too big, the downcomer liquid seal will be displaced by the vapor from the tray below, which will displace the liquid in the downcomer, which will cause the downcomer and the trays above to flood.

My standard for setting the downcomer clearance is then:

1. About $\frac{1}{4}$ - to $\frac{1}{2}$ -inch below the top of the outlet weir. Some engineers will rely on a dynamic seal, meaning they count on the crest height of the liquid overflowing the outlet weir to submerge the bottom edge of the downcomer from the tray above. Not me. I worry too much about tray deck leakage, liquid turndown, and out-of-level weirs. So I'll stick with the $\frac{1}{4}$ -inch minimum overlap that I've relied on for 47 years.
2. My minimum downcomer clearance is 1 inch. Maybe a pressure surge will push up the downpour plate below the downcomer? Maybe corrosion deposits will accumulate in this area?
3. I maintain a minimum design horizontal velocity (called the escape velocity) of about 1 to $1\frac{1}{2}$ ft per second below the bottom edge of the downcomer to prevent vapor intrusion into the downcomer (See V_1 , Fig. 5.4).

Of course, these sorts of design dimensions are meaningless unless both the weirs and bottom edge of the downcomers are level. So inspect the installed trays for weir and downcomer levelness.

5.1.12 Inlet Weirs

There's a problem with my design criteria for downcomer clearances. That is, at low liquid rates, how can I keep a reasonable downcomer escape velocity without reducing the downcomer velocity below my 1-inch clearance? The conventional answer is an inlet weir, as shown in Fig. 5.5.

But there's a nasty potential problem with inlet weirs. If downcomer A bulges out (see Fig. 5.5), then the gap shown as distance X between the inlet weir, which seals the downcomer, and the downcomer itself will be reduced. As I've seen before, when this happens, tower flooding results. So maintaining the gap shown as distance X in Fig. 5.5 is very important.

I have used inlet weirs to guarantee a downcomer seal at low liquid rates. But I'll leave a generous horizontal space (maybe 3 or 4 inches) between the inlet weir and downcomer A in Fig. 5.5 (i.e., distance X could be 3 or 4 inches).

Also, I'll inspect the installed trays to assure myself that the downcomer itself is not too flexible at its bottom edge. This is a good application for downcomer bracing brackets.

Finally, the inlet weir is also a dirt trap (like the recessed sump). So I would be afraid to use it in fouling or corrosive services.

5.1.13 Seal Pan

Below the downcomer of the bottom tray is the seal pan. The seal pan, like a recessed sump, is also a potential dirt trap. I've crawled

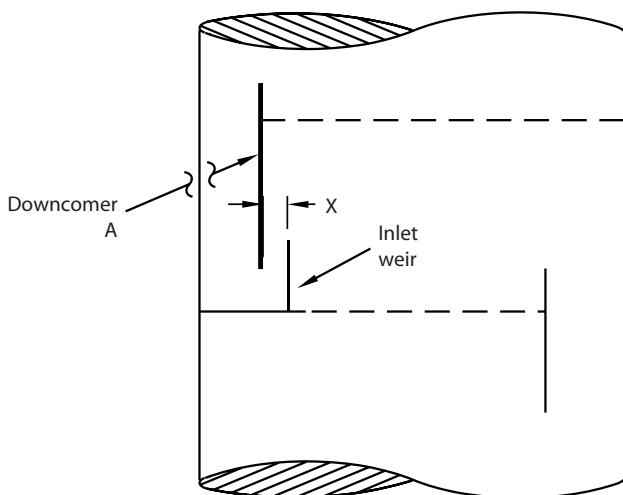


FIGURE 5.5 Inlet weir provides a positive downcomer seal.

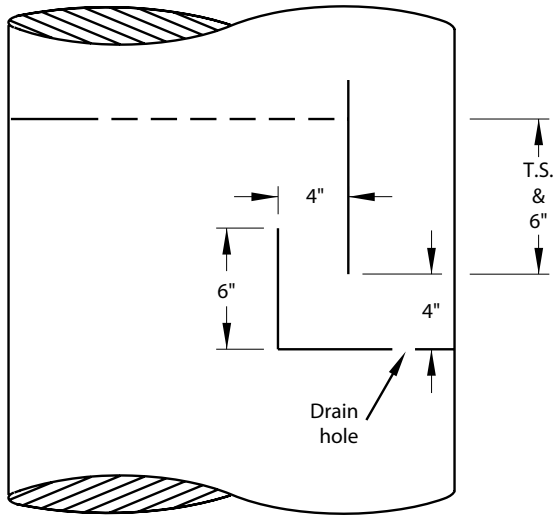


FIGURE 5.6 Seal pan with drain hole.

through many towers and have seen the seal pans half full of dirt. So I have developed the following design criteria (Fig. 5.6):

1. Clearances between the seal pan and downcomer are double normal those that would be used for the downcomer and the tray deck.
2. The downcomer itself is longer than normal.
3. A drain hole is provided in the floor of the seal pan, and is sized for 20–25 percent of the design flow, to flush out the accumulated dirt.

To calculate the size of this drain hole, first calculate the drain hole velocity, V :

$$H = 0.6 \cdot V \text{ (feet per second)}$$

The H factor is the depth of the seal pan (shown in Fig. 5.6) plus about 1 inch to allow for the crest height over the top edge of the seal pan.

Multiply the design volumetric flow by 25 percent and divide by V to determine the hole area in square feet.

Of course, if you oversize the seal pan drain hole, you will lose the downcomer seal and flood the tower. A drain hole of less than $\frac{1}{4}$ inch will likely plug.

Before closing up a tower, don't neglect to inspect the seal pan to make sure it's free of gloves and dead rats.

5.1.14 Downcomer Bracing Brackets

In small-diameter towers using single-pass trays, this is not an issue. But, once a tower is larger than 8 ft in diameter or multipass trays are used, downcomer bracing brackets are critical.

These brackets are attached to the bottom edge of the side downcomers to keep the downcomer bottom edge from flexing. For example, pressure below a tray can force the downcomer bottom area to be reduced and to restrict liquid flow. This may cause the downcomer to fill and flood. I like to keep the bracket spacing to less than 3 ft. For two-pass trays, the center downcomer spacing is maintained by a ½-inch hollow cylinder bolted between the opposite walls of the center downcomer, also with 3-ft spacing.

Failure to secure the side downcomer bracing brackets is a common installation error. So one needs to specify the spacing correctly, and then check the installation in the column.

5.1.15 Selection of Tray Caps

Here's my advice on this critical subject:

1. Don't use movable caps (i.e., valves). They do not greatly decrease tray deck leakage at low vapor rates. But they do stick to the tray deck and cause premature flooding.
2. For clean services, I use Sulzer MVG grid trays.
3. For fouling services, I use Koch-Glitsch pro-valve grid trays with a ½-inch lift.
4. If capacity and vapor flow variability (i.e., turndown) is not an issue, I use sieve trays with 3/8-inch holes (clean service) or ½-inch holes (dirty services). Sieve trays have about 10 percent less capacity than grid trays. But anyone can make them in a day (i.e., new tray panels) and the price is minimal.

5.1.16 Materials of Construction

Only fools use thin, carbon steel trays:

1. The industry standard is 14 gauge. My standard is 10 gauge, or 8 gauge for vacuum tower steam stripping trays. The smaller the gauge, the thicker the metal.
2. In virgin crude oil service I use 316 S.S. for naphthenic acid protection, which requires moly and nickel.
3. In thermally cracked service, where naphthenic acids have been destroyed, I use 410 steel.

After 47 years of practice, I have never used carbon steel because as it corrodes, it chokes off the lift area underneath the cap and leads to higher vapor ΔP and flooding.

Make the downcomers thick too. Flexibility in downcomers is bad. They need to be kept stiff and strong.

5.1.17 Tray Mechanical Details

I have shown detailed drawings of the following features in my book, *Process Design for Reliable Operations*, available on Amazon (make sure you get the third edition):

1. Back-to-back trays: See Fig. 5.7.
2. Shear clips: Weld the shear clip to the underside of the tray as shown in Fig. 5.8, but certainly not to the tray ring.
3. Bolted-in trays: Use the slotted or oversized holes on the trays as a guide to drill or burn holes in the tray rings. Then bolt in the trays on 4- or 6-inch spacing. Use a double-nutting arrangement. Secure the first nut finger tight and $\frac{1}{4}$ turn with

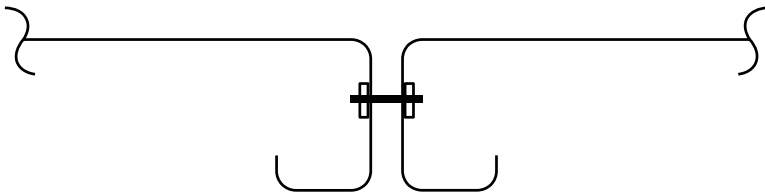


FIGURE 5.7 Back-to-back tray.

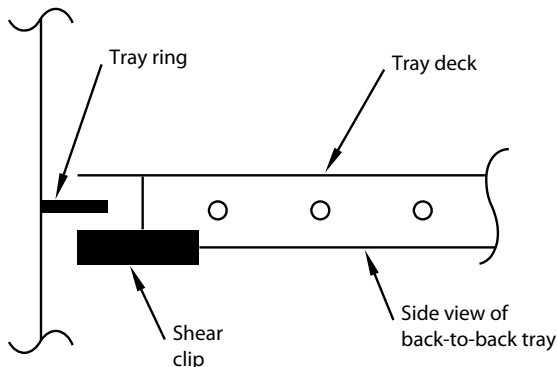


FIGURE 5.8 Shear clip design.

a wrench. This permits tray movement as the tray heats up. The backup nut is tightened up against the first nut to keep the first nut from coming loose.

4. Explosion doors: These are now standard in the industry. Use mechanical door stops to close the doors, rather than chains, which tend to break.
5. Use grid trays: They are stronger than sieve or valve trays. Again, I use 10 gauge rather than the standard 14-gauge deck thickness.

CHAPTER 6

Why Control Tower Pressure

Options for Optimizing Tower Operating Pressure

Why are distillation towers designed with controls that fix the tower pressure?

Naturally, we do not want to overpressure the tower and pop open the safety relief valve. Alternatively, if the tower pressure gets too low, we could not condense the reflux. Then the liquid level in the reflux drum would fall and the reflux pump would lose suction and cavitate. But assuming that we have plenty of condensing capacity and are operating well below the relief valve set pressure, why do we attempt to fix the tower pressure? Further, how do we know what pressure target to select?

I well remember one pentane-hexane splitter in Toronto. The tower simply could not make a decent split, regardless of the feed or reflux rate selected. The tower-top pressure was swinging between 12 and 20 psig. The flooded condenser pressure control valve, shown in Fig. 6.1, was operating between 5 and 15 percent open, and hence it was responding in a non-linear fashion (most control valves work properly only at 20 to 75 percent open). The problem may be explained as follows.

The liquid on the tray deck was at its bubble, or boiling, point. A sudden decrease in the tower pressure caused the liquid to boil violently. The resulting surge in vapor flow promoted jet entrainment, or flooding.

Alternately, the vapor flowing between trays was at its dew point. A sudden increase in tower pressure caused a rapid condensation of this vapor and a loss in vapor velocity through the tray deck holes. The resulting loss in vapor flow caused the tray decks to dump.

Either way, erratic tower pressure results in alternating flooding and dumping, and therefore reduced tray efficiency. While gradual

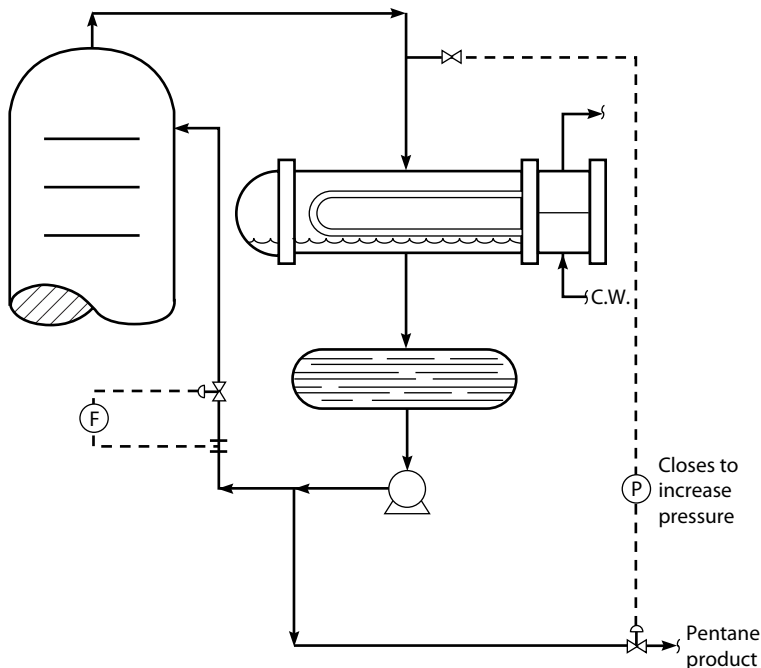


FIGURE 6.1 Flooded condenser pressure control.

swings in pressure are quite acceptable, no tower can be expected to make a decent split with a rapidly fluctuating pressure.

6.1 Selecting an Optimum Tower Pressure

The process design engineer selects the tower design operating pressure as follows:

1. Determines the maximum cooling water or ambient air temperature that is typically expected on a hot summer day in the locale where the plant is to be built.
2. Calculates the condenser outlet, or reflux drum temperature, that would result from the above water or air temperature (as discussed in Chap. 18).
3. Referring to Fig. 6.2, the designer calculates the pressure in the reflux drum, assuming that the condensed liquid is at its bubble point (as discussed in Chap. 14). Adding 5 or 10 psig to this pressure for pressure loss in the overhead condenser and associated piping, the designer then determines the tower-top pressure.

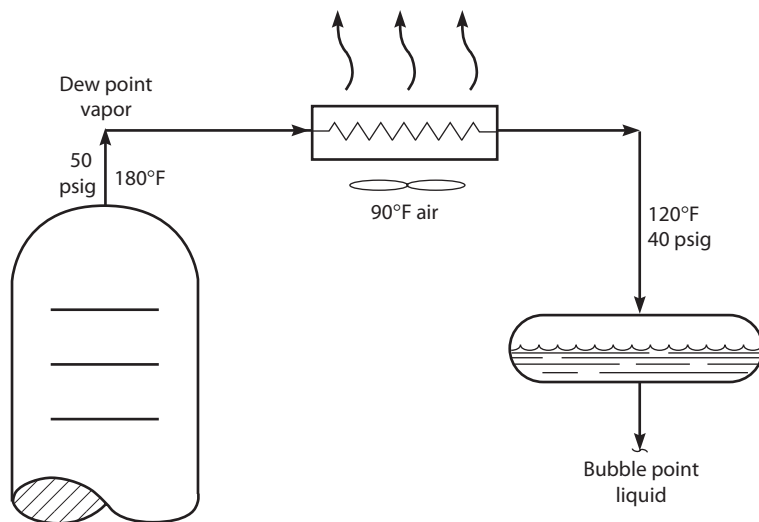


FIGURE 6.2 Calculating the tower design pressure.

Of course, the unit operator can physically deviate from this design pressure, but to what purpose?

6.2 Raising the Tower Pressure Target

I once had a contract in a Denver refinery to revamp a hydrocracker fractionator, which produced naphtha, jet fuel, and diesel. The bottle-neck was tray flooding. At higher feed rates, the kerosene would carry over into the overhead naphtha product. My initial plant inspection showed that the tower-top pressure was 24 psig. The relief valve was set at 50 psig. By raising the tower operating pressure to 30 psig, the flooding was stopped and my contract was canceled. Why?

Ambient pressure in Denver is about 13 psia (lb/in² absolute) (vs. 14.7 psia at sea level). Higher pressures reduce the volumetric flow of vapor. In other words, volume is inversely proportional to pressure:

$$\text{Volume} \sim \frac{1}{\text{pressure}}$$

The pressure we are concerned with is the absolute pressure:

$$\text{Initial pressure} = 24 \text{ psig} + 13 \text{ psi} = 37 \text{ psia}$$

$$\text{Final pressure} = 30 \text{ psig} + 13 \text{ psi} = 43 \text{ psia}$$

The absolute tower pressure (in psia) increased by 17 percent, and hence the volume (as well as the velocity of vapor through the valve

tray caps) declined by 17 percent. The reduced vapor velocity reduced the dry tray pressure drop, thus reducing both the spray height above the tray deck and the liquid backup in the downcomers.

Another reason to raise tower pressure is to permit higher reflux rates. If the pressure controller in Fig. 6.1 is set too low, then during hot weather, when condenser capacity becomes marginal, the level in the reflux drum will be lost. If we then raise the pressure set point, the drum will refill—but why?

Raising the tower pressure also increases the reflux drum pressure, raising, in turn, the temperature at which the vapors condense. The rate of condensation is then calculated from the following:

$$Q = U \times A (T_C - T_A)$$

where Q = rate of condensation, Btu/h

A = heat-exchanger surface area, ft²

U = heat-transfer coefficient, Btu/[$(h)(ft^2)(^\circ F)$]

T_C = condensation temperature of vapors, $^\circ F$

T_A = temperature of air or cooling water, $^\circ F$

Raising the tower pressure increases T_C .

6.3 Lowering the Tower Pressure

In general, distillation columns should be operated at a low pressure. For example, Fig. 6.3 shows an isobutane-normal butane stripper. This fractionator is performing poorly. A computer simulation of the column has been built. The column has 50 actual trays. But in order to force the computer model to match existing operating parameters (reflux rate, product compositions), 10 theoretical separation stages (i.e., 10 trays, each 100 percent efficient) must be used in the model. This means that the trays are developing an actual tray efficiency of only 20 percent.

A field measurement indicated a pressure drop of 2.0 psi. Assuming a specific gravity of 0.50, then the pressure drop per tray in inches of liquid is:

$$\begin{aligned} \frac{2.0 \text{ psi tower } \Delta P}{50 \text{ tray}} \times \frac{28 \text{ inch H}_2\text{O}}{1 \text{ psi}} \times \frac{1.0 \text{ (s.g. water)}}{0.50 \text{ (s.g. Butane)}} \\ = 2.24 \text{ inch liquid } \Delta P \text{ per tray} \end{aligned}$$

As the weir height of the trays is 3 inch, it is a safe assumption that the low tray efficiency is due to tray deck dumping, rather than flooding. As shown in Fig. 6.3, this column has no reflux. This is a typical design for strippers; when feed is introduced on the top tray, there is no need for reflux.

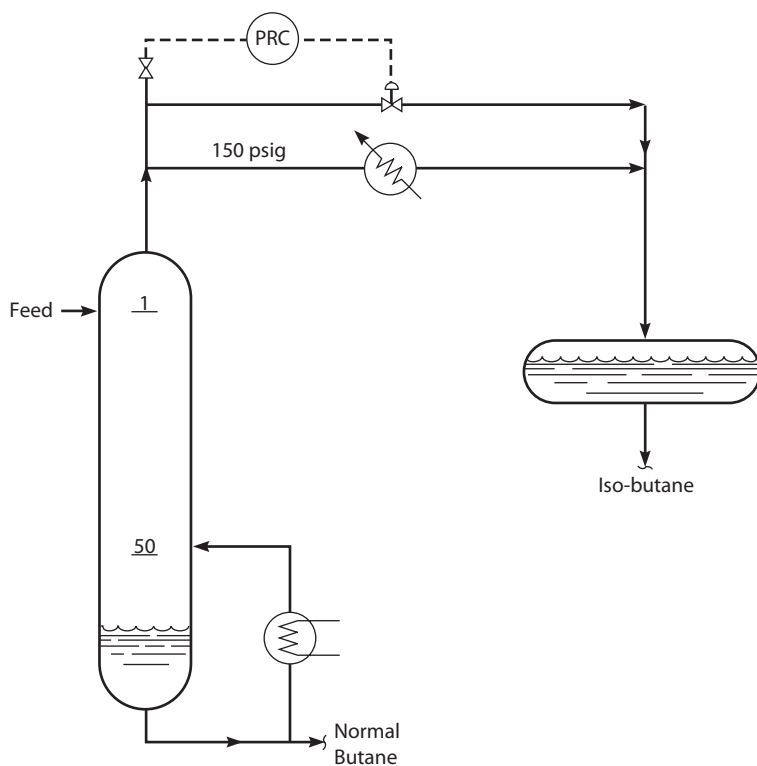


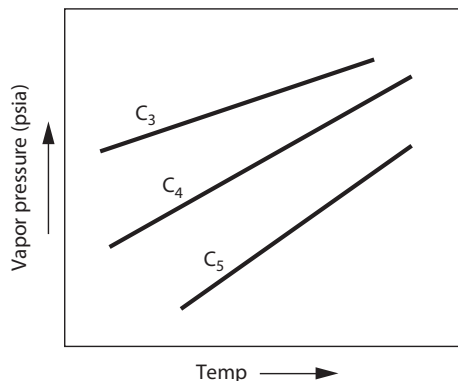
FIGURE 6.3 Isobutane stripper with hot-vapor bypass pressure controller.

In order to improve tray efficiency, it will be necessary to increase the vapor velocity through the trays, so as to increase the pressure drop to at least 4 or 5 inch of liquid per tray. If the reboiler duty were simply increased, the concentration of the heavy component—normal butane—in the light overhead product—isobutane—would escalate exponentially. Another method, however, that does not involve increasing either the reboiler duty or the *mass flow* of vapor through the trays can be used to increase vapor velocity.

By lowering the tower operating pressure, the volume of vapor flow bubbling up through the tray decks may be increased without changing the mass flow. For instance, if the tower pressure were reduced from 105 psig (or 120 psia) to 45 psig (or 60 psia), then the velocity of vapor through the sieve holes on the trays would double. This would lead to a substantial increase in the dry tray pressure drop, and hence reduce tray deck leakage.

To lower the tower pressure, the hot-vapor bypass pressure recorder controller (PRC) valve is closed. This forces more vapor through the condenser, which, in turn, lowers the temperature in the

FIGURE 6.4
Vapor-pressure
chart.



reflux drum. As the liquid in the reflux drum is at its bubble point, reducing the reflux drum temperature will reduce the reflux drum pressure. As the stripper tower pressure floats on the reflux drum pressure, the pressure in the tower will also decline.

The net effect of reducing the stripper pressure was to greatly reduce the amount of isobutane in the heavier normal butane bottoms product. Undoubtedly, most of the improvement in fractionation was due to enhanced tray efficiency, which resulted from suppressing tray deck leaking, or dumping. But there was a secondary benefit of reducing tower pressure: increased *relative volatility*.

6.3.1 Relative Volatility

The chart shown in Fig. 6.4 is called a *Cox*, or *vapor-pressure*, chart. It shows the pressure developed by pure-component liquids at various temperatures. The interesting aspect of this chart is that the sloped lines, representing the vapor pressures of pure hydrocarbon components, spread apart at lower pressures. This results in an increase in the ratio of the vapor pressures of any two components.

The vapor pressure of a light component at a given temperature divided by the vapor pressure of a heavier component at the same temperature is called the *relative volatility*. For practice, calculate the relative volatility of isobutane and normal pentane at 140°F (answer: 4.0). Next, calculate their relative volatility at 110°F (answer: 4.9).*

Note that the relative volatility has increased by about 20 percent at the lower temperature and pressure. This increase in relative volatility allows one to make a better split at a given reflux rate, or to

*You will need to use a vapor-pressure chart for this calculation. One source is your *API Data Book* (American Petroleum Institute).

make the same split at a lower reflux rate. We can quantify this last statement as follows:

$$\frac{(RVL - RVH)}{(RVH - 1)} = DRF$$

where RVH = relative volatility at a high pressure

RVL = relative volatility at a low pressure

DRF = percent reduction in the reflux rate, when the same degree of fractionation is desired

Reducing reflux saves reboiler duty. Also, the lower pressure will reduce the tower-bottom temperature, and this also cuts the reboiler energy requirement. For most distillation towers, the energy cost of the reboiler duty is the main component of the total operating cost to run the tower.

6.3.2 Incipient Flood Point

As an operator reduces the tower pressure, three effects occur simultaneously:

- Relative volatility increases.
- Tray deck leakage decreases.
- Entrainment, or spray height, increases.

The first two factors help make fractionation better, the last factor makes fractionation worse. How can an operator select the optimum tower pressure to maximize the benefits of enhanced relative volatility, and reduced tray deck dumping without unduly promoting jet flooding due to entrainment?

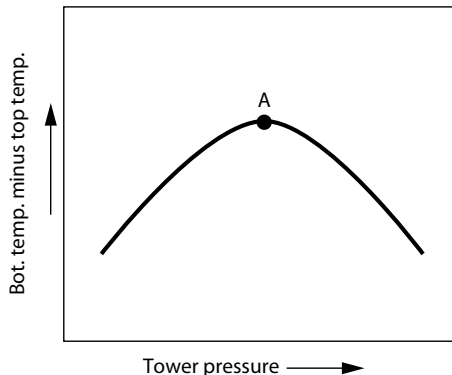
To answer this fundamental question, we should realize that reducing the tower pressure will also reduce both the tower-top temperature and the tower-bottom temperature. So the change in these temperatures, by themselves, is not particularly informative. But if we look at the difference between the bottom and top temperatures, this difference is an excellent indication of fractionation efficiency. The bigger this temperature difference, the better the split. For instance, if the tower-top and tower-bottom temperatures are the same for a 25-tray tower, what is the average tray efficiency? (Answer: $100 \text{ percent} \div 25 = 4 \text{ percent}$.)

The basis for the 100 percent is that if the tower top and bottom temperatures are identical (plus or minus a few degrees), then the entire tower is functioning as if it represented a single perfect theoretical fractionation stage. The fractionation efficiency of a theoretical stage is equal to one perfect tray operating at 100-percent efficiency.

Figure 6.5 illustrates this relationship. Point A is the incipient flood point. In this case, the *incipient flood point* is defined as the

FIGURE 6.5

Point A represents the optimum tower pressure.



operating pressure that maximizes the temperature difference across the tower at a particular reflux rate. How, then, do we select the optimum tower pressure to obtain the best efficiency point for the trays? (Answer: Look at the temperature profile across the column.)

6.4 The Phase Rule in Distillation

This is perhaps an idea you remember from high school, but never quite understood. The phase rule corresponds to determining how many independent variables we can fix in a process before all the other variables become *dependent variables*. In a reflux drum, we can fix the temperature and composition of the liquid in the drum. The temperature and composition are called *independent variables*. The pressure in the drum could now be calculated from the chart in Fig. 6.4. The pressure is a dependent variable. The *phase rule* for the reflux drum system states that we can select any two variables arbitrarily (temperature, pressure, or composition), but then the remaining variable is fixed.

A simple distillation tower, like that shown in Fig. 6.2, also must obey its own phase rule. Here, because the distillation tower is a more complex system than the reflux drum, there are three independent variables that must be specified. The operator can choose from a large number of variables, but must select no more than three from the following list:

- Tower pressure
- Reflux rate, or reflux ratio
- Reboiler duty
- Tower-top temperature
- Tower-bottom temperature
- Overhead product rate

- Bottoms product rate
- Overhead product composition
- Bottoms product composition

The prior discussion assumes that the feed rate, feed composition, and heat content of the feed (enthalpy) are fixed. My purpose in presenting this review of the phase rule is to encourage the routine manipulation of tower-operating pressures in the same sense, and with the same objectives, as adjusting reflux rates. Operators who arbitrarily run a column at a fixed tower pressure are discarding one-third of the flexibility available to them to operate the column in the most efficient fashion. And this is true regardless of whether the objective is to save energy or improve the product split.

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CHAPTER 7

What Drives Distillation Towers

Reboiler Function

An internal-combustion engine drives a car. Pumps are driven by turbines or motors. Jet planes are pushed by the thrust of an axial air compressor.

7.1 The Reboiler

All machines have drivers. A distillation column is also a machine, driven by a *reboiler*. It is the heat duty of the reboiler, supplemented by the heat content (enthalpy) of the feed, that provides the energy to make a split between light and heavy components. A useful example of the importance of the reboiler in distillation comes from the venerable use of sugarcane in my home state of Louisiana.

If the cut cane is left in the fields for a few months, its sugar content ferments to alcohol. Squeezing the cane then produces a rather low-proof alcoholic drink. Of course, one would naturally wish to concentrate the alcohol content by distillation, in the still shown in Fig. 7.1.

The alcohol is called the light component, because it boils at a lower temperature than water; the water is called the heavy component, because it boils at a higher temperature than alcohol. Raising the top reflux rate will lower the tower-top temperature and reduce the amount of the heavier component, water, in the overhead alcohol product. But what happens to the weight of vapor flowing up through the trays? Does the flow go up, go down, or remain the same?

There are two ways to answer this question. Let's first look at the reboiler. As the tower-top temperature shown in Fig. 7.1 goes down, more of the lighter, lower-boiling-point alcohol is refluxed down the tower. The tower-bottom temperature begins to drop, and the steam flow to the reboiler is automatically increased by the action of the

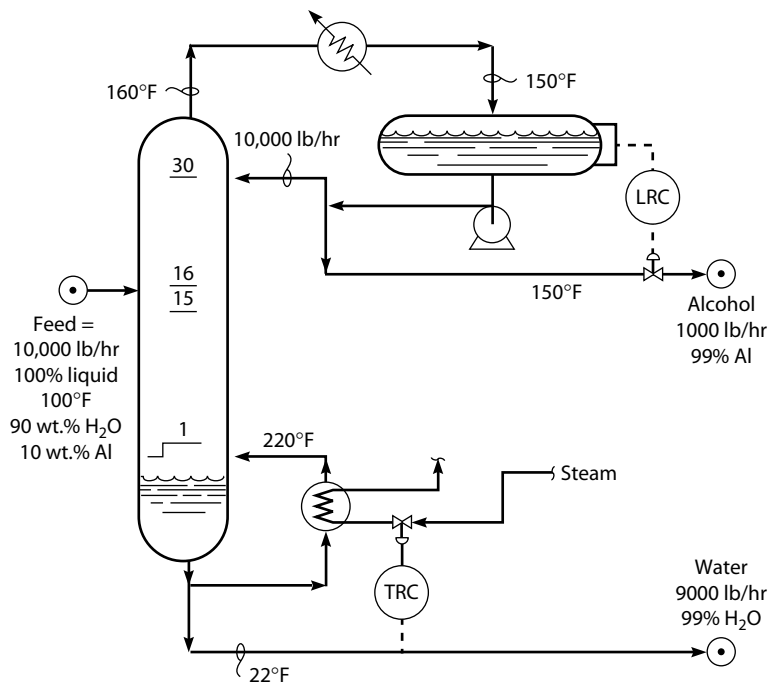


FIGURE 7.1 Alcohol-water splitter.

temperature recorder controller (TRC). As the steam flow to the reboiler increases, so does the reboiler duty (or energy injected into the tower in the form of heat). Almost all the reboiler heat or duty is converted to vaporization. We will prove this statement mathematically later in this chapter. The increased vapor leaving the reboiler then bubbles up through the trays, and hence the flow of vapor is seen to increase as the reflux rate is raised.

Now let's look at the reflux drum. The incremental reflux flow comes from this drum. But the liquid in this drum comes from the condenser. The feed to the condenser is vapor from the top of the tower. Hence, as we increase the reflux flow, the vapor rate from the top of the tower must increase. One way of summarizing these results is to say that the *reflux comes from the reboiler*.

The statement that the mass, or weight flow of vapor through the mass, increases as the reflux rate is raised is based on the reboiler being on automatic temperature control. If the reboiler were on manual control, then the flow of steam and the reboiler heat duty would remain constant as the reflux rate was increased, and the weight flow of vapor up the tower would remain constant as the top reflux rate was increased. But the liquid level in the reflux drum would begin to drop. The reflux drum level recorder controller (LRC) would close off

to catch the falling level, and the overhead product rate would drop in proportion to the increase in reflux rate. We can now draw some conclusions from the foregoing discussion:

- The flow of vapor leaving the top tray of the tower is equal to the flow of reflux, plus the flow of the alcohol overhead product.
- The overhead condenser heat-removal duty is proportional to the reboiler heat duty.
- The weight flow of vapor in a tower is controlled by one factor and one factor only: *heat*.

An increase in reflux rate, assuming that the reboiler is on automatic temperature control, increases both the tray weir loading and the vapor velocity through the tray deck. This increases both the total tray pressure drop and the height of liquid in the tray's downcomer. Increasing reflux rates, with the reboiler on automatic temperature control, will always push the tray closer to or even beyond the point of incipient flood.

7.2 Heat-Balance Calculations

If you have read this far and understood what you have read, you will readily understand the following calculation. It is simply a repetition, with numbers, of the discussion previously presented. However, you will require the following values to perform the calculations:

- Latent heat of condensation of alcohol vapors = 400 Btu/lb
- Latent heat of condensation of water vapors = 1000 Btu/lb
- Specific heat of alcohol (vapor or liquid) = $0.6 \text{ Btu}/[(\text{lb})(^\circ\text{F})]$
- Specific heat of water = $1.0 \text{ Btu}/[(\text{lb})(^\circ\text{F})]$

The term *specific heat* refers to the sensible-heat content of either vapor or liquid. The specific heat is the amount of heat needed to raise the temperature on one pound of the vapor or liquid by 1°F . The term *latent heat* refers to the heat of vaporization, or the heat of condensation, needed to vaporize or condense one pound of liquid or vapor at constant temperature. Note that the heat of condensation is equal to the heat of vaporization. Each is referred to as the *latent heat*. The sum of the sensible heat, plus the latent heat, is called the *total heat content*, or *enthalpy*.

Returning to our example in Fig. 7.1, we wish first to determine the reboiler duty. To do this, we have to supply three heat requirements:

- A. Heat 9000 lb/h of water from the 100°F feed temperature to the tower-bottom temperature of 220°F .

- B. Heat 1000 lb/h of alcohol from the 100°F feed temperature (where the alcohol is a liquid) to the tower overhead temperature of 160°F (where the alcohol is a vapor).
- C. Vaporize 10,000 lb/h of reflux from the 150°F reflux drum temperature to the tower overhead temperature of 160°F.

Solution to step A:

$$9000 \text{ lb/h} \times 1.0 \text{ Btu}/[(\text{lb})(^\circ\text{F})] \times (220^\circ\text{F} - 100^\circ\text{F}) = 1,080,000 \text{ Btu/h}$$

Solution to step B:

$$1000 \text{ lb/h} \times 0.6 \text{ Btu}/[(\text{lb})(^\circ\text{F})] \times (160^\circ\text{F} - 100^\circ\text{F}) + 1000 \text{ lb/h} \\ \times 400 \text{ Btu/lb} = 36,000 \text{ Btu/h} + 400,000 \text{ Btu/h} = 436,000 \text{ Btu/h}$$

Solution to step C:

$$10,000 \text{ lb/h} \times 0.6 \text{ Btu}/[(\text{lb})(^\circ\text{F})] \times (160^\circ\text{F} - 150^\circ\text{F}) + 10,000 \text{ lb/h} \\ \times 400 \text{ Btu/lb} \\ = 60,000 \text{ Btu/h} + 4,000,000 \text{ Btu/h} = 4,060,000 \text{ Btu/h}$$

The reboiler duty is then the sum of A + B + C = 5,576,000 Btu/h.

The next part of the problem is to determine the vapor flow to the bottom tray. If we assume that the vapor leaving the reboiler is essentially steam, then the latent heat of condensation of this vapor is 1000 Btu/lb. Hence the flow of vapor (all steam) to the bottom tray is

$$= 5,576,000 \text{ Btu/h} \div 1000 \text{ Btu/lb} = 5576 \text{ lb/h}$$

What about the vapor flow leaving the top tray of our splitter? That is simply the sum of the reflux plus the overhead product:

$$10,000 \text{ lb/h} + 1000 \text{ lb/h} = 11,000 \text{ lb/h}$$

How about the condenser duty? That is calculated as follows:

$$11,000 \text{ lb/h} \times 0.6 \text{ Btu}/[(\text{lb})(^\circ\text{F})] \times (160^\circ\text{F} - 150^\circ\text{F}) \\ + 11,000 \text{ lb/h} \times 400 \text{ BTU/lb} \\ = 66,000 \text{ BTU/h} + 4,400,000 \text{ BTU/h} = 4,466,000 \text{ BTU/h}$$

We can draw the following conclusions from this example:

- The condenser duty is usually a little smaller than the reboiler duty.
- Most of the reboiler heat duty usually goes into generating reflux.
- The flow of vapor up the tower is created by the reboiler.

For other applications, these statements may be less appropriate. This is especially so when the reflux rate is much smaller than the

feed rate. But if you can grasp these calculations, then you can appreciate the concept of the reboiler acting as the engine to drive the distillation column.

7.2.1 Effect of Feed Preheat

Up to this point, we have suggested that the weight flow of vapor up the tower is a function of the reboiler duty only. Certainly, this cannot be completely true. If we look at Fig. 7.2, it certainly seems that increasing the heat duty on the *feed preheater* will reduce the reboiler duty.

Let us assume that both the reflux rate and the overhead propane product rate are constant. This means that the total heat flow into the tower is constant. Or the sum of the reboiler duty plus the feed preheater duty is constant. If the steam flow to the feed preheater is increased, then it follows that the reboiler duty will fall. How does this increase in feed preheat affect the flow of vapor through the trays and the fractionation efficiency of the trays?

The bottom part of the tower in Fig. 7.2—that is, the portion below the feed inlet—is called the *stripping section*. The upper part of

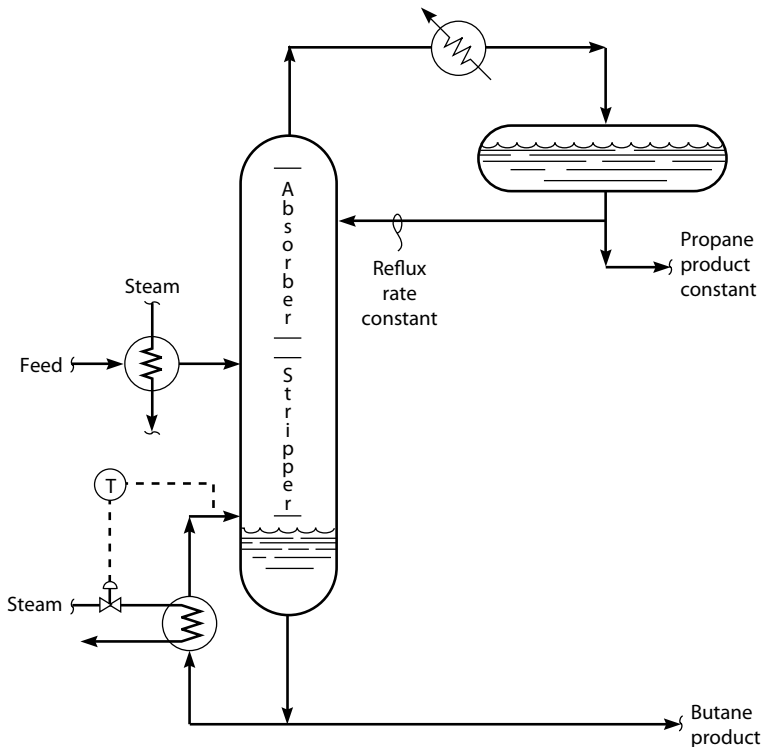


FIGURE 7.2 Feed preheat affects stripping efficiency.

the tower—that is, the portion above the feed inlet—is called the *absorption section*.

Since both the reflux flow and the overhead product flow are constant in this problem, it follows that the weight flow of vapor leaving the top tray is also constant, regardless of the feed preheater duty. Actually, this statement is approximately true for all the trays in the top or absorption part of the tower. Another way of saying this is that the heat input to the tower above the feed tray is a constant.

But for the bottom stripping section trays, a reduction in reboiler duty will directly reduce the vapor flow from the reboiler to the bottom tray. This statement is approximately valid for all the trays in the stripping section of the tower.

As the flow of vapor through the absorption section trays is unaffected by feed preheat, the fractionation efficiency of the trays in the upper part of the tower will not change as feed preheat is increased. On the other hand, the reduced vapor flow through the stripping section may increase or decrease fractionation efficiency—but why?

7.2.2 Optimizing Feed Preheater Duty

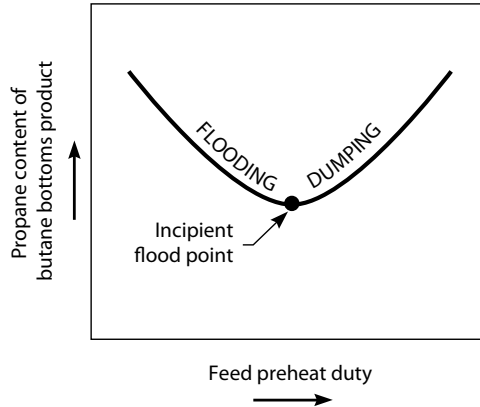
Trays suffer from lost tray efficiency as a result of both flooding and dumping. Trays have some entrained droplets of liquid lifted by the flowing vapors to the trays above. This tends to blow butane up into the lighter propane product. Perforated trays always have some leakage of liquid through the tray deck to the trays below. This tends to drip propane down into the heavier butane product.

When we increase feed preheat and the reboiler duty is automatically reduced, dumping increases, but entrainment decreases. If the trays below the feed point were working poorly because they were flooding, increasing feed preheat would improve their fractionation efficiency. If the trays below the feed point were working poorly because they were dumping, increasing feed preheat would reduce their fractionation efficiency. Figure 7.3 summarizes this effect. If, for this tower, we arbitrarily state that the percent of butane in the overhead propane product is constant, then the feed preheat duty, which minimizes the propane content in the butane bottoms product, represents the optimum preheater duty. This preheater duty corresponds to the *incipient flood point*. The optimum feed preheater duty maximizes fractionation at a fixed reflux rate.

Varying the heat content of the feed is an additional independent variable that an operator can use to optimize fractionation efficiency. An additional benefit of feed preheat is that a lower-level temperature heat source can be used. If valuable 100-psig steam is required for the reboiler, then low-value 20-psig steam might be adequate for the feed preheat exchanger.

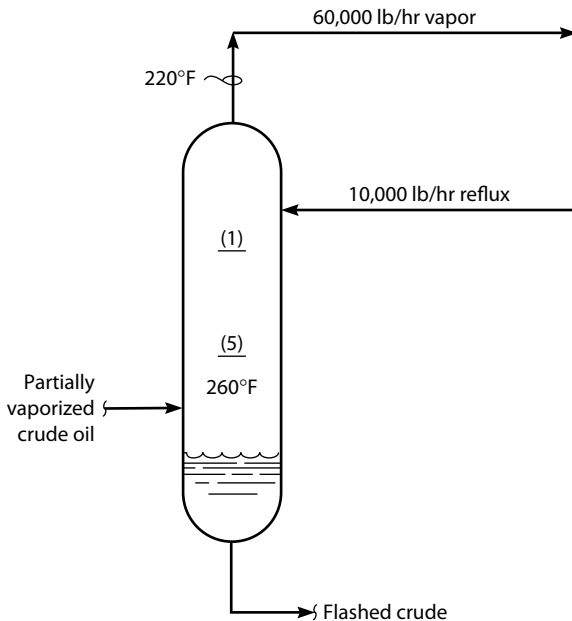
FIGURE 7.3

Optimizing feed preheat duty at a constant reflux rate.



7.2.3 Multicomponent Systems

So far, all our examples have dealt with two-component systems, and many of our towers really just have two components. Also, we have assumed that the reflux rate is large compared to the overhead product rate, and many of our towers do run with a lot of reflux. But we can all think of distillation columns where the top reflux rate is small compared to the overhead product, and the overhead product itself consists of a dozen widely different chemical compounds. Figure 7.4 represents such a column. It is called a *crude preflash tower*.

**FIGURE 7.4** Crude preflash tower.

Notice that there is no reboiler in this flash tower. All the heat input comes from the partially vaporized crude. Both the temperature and the percent vaporization of the crude are fixed. Hence, the *external* heat input to the preflash tower is constant. The pounds of vapor flowing to the bottom tray must also be constant.

Now the overhead product of this tower is a mixture of a hundred different components, ranging from ethane, which has a molecular weight of 30, to decane, which has a molecular weight of 142. Also, while the overhead vapor rate is 60,000 lb/h, the top reflux rate is only 10,000 lb/h.

Consider the following. When the operator raises the top reflux rate, what happens to the weight flow of vapor going to the top tray? Recalling that the external heat input to this tower is constant, do the pounds per hour of vapor flowing to the top tray increase, remain the same, or decrease? The correct answer is *increase*. But why?

7.2.4 Conversion of Sensible Heat to Latent Heat

When we raise the top reflux rate to our preflash tower, the tower-top temperature goes down. This is a sign that we are washing out from the up-flowing vapors more of the heavier or higher-molecular-weight components in the overhead product. Of course, that is why we raised the reflux rate. So the reduction in tower-top temperature is good.

But what happened to the sensible-heat content (the heat represented by the temperature) of the vapors leaving the tower? As the vapor is cooler, the sensible-heat content decreased. Where did this heat go?

A small part of the heat was picked up by the extra liquid draining from the top tray. This extra liquid comes from the extra reflux. But the liquid flow through the tower is too small to carry away much heat. The main reason why the vapors leaving the top tray are cooler is vaporization; in other words, the sensible-heat content of the flowing vapors is converted to latent heat of vaporization.

But what is vaporizing? The reflux, of course. The sensible-heat content of the vapors, which is reduced when the reflux rate is increased, is converted to latent heat as the vapors partially vaporize the incremental reflux flow.

As the reflux rate is raised, the weight flow of vapor through the top tray, and to a lesser extent through all the trays below (except for the bottom tray), increases. This increase in the weight flow of vapor occurs even though the external heat input to the preflash tower is constant. The weight flow of vapor to the bottom tray is presumed to be solely a function of the pounds of vapor in the feed.

7.2.5 Reduced Molecular Weight

A reduction in tower-top temperature of 20°F would increase the weight flow of vapor by roughly 10 percent. But the composition of

the vapor would also change. The molecular weight of the vapor would drop by approximately 8 percent. The lower the molecular weight (MW) of a gas, the greater the volume that a given weight of the gas occupies.

$$\text{Gas volume} \approx \frac{\text{weight of gas}}{\text{MW of gas}}$$

In this equation, if the weight of gas goes up by 10 percent, and the molecular weight of the gas goes down by 8 percent, then the volume of gas goes up by 18 percent. The reduction in the tower-top temperature of 20°F does shrink the gas by about 3 percent, as a result of the temperature reduction, so that the net effect of raising the reflux rate is to increase the gas volume through the top tray by 15 percent (i.e., 18 percent minus 3 percent). This results in a substantial increase in the top tray pressure drop, which can, and often does, cause the top tray to flood. This can happen even though the external heat input and feed rate to the tower have never changed.

7.2.6 Internal Reflux Evaporation

The tray temperatures in our preflash tower, shown in Fig. 7.4, drop as the gas flows up the tower. Most of the reduced sensible-heat content of the flowing gas is converted to latent heat of evaporation of the downflowing reflux. This means that the liquid flow, or internal reflux rate, decreases as the liquid flows down the column. The greater the temperature drop per tray, the greater the evaporation of internal reflux. It is not unusual for 80 to 90 percent of the reflux to evaporate between the top and bottom trays in the absorption section of many towers. We say that the lower trays in the absorption section of such a tower are “drying out.” The separation efficiency of trays operating with extremely low liquid flows over their weirs will be very low. This problem is commonly encountered for towers with low reflux ratios and a multicomponent overhead product composition.

7.2.7 Overview

I once claimed that I could teach any interested, reasonably intelligent person the elements of chemical engineering in two hours. The part of this chapter about the heat balance calculations represents my effort to defend this claim. If you are not a chemical engineer and you have understood the preceding calculations, then you may know enough to pass as a degreed chemical engineer. Well, maybe I’m exaggerating. Maybe you had better read the rest of this book first.

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CHAPTER 8

How Reboilers Work

Thermosyphon, Gravity Feed, and Forced

Four types of reboilers are discussed in this chapter:

- Once-through thermosyphon reboilers
- Circulating thermosyphon reboilers
- Forced-circulation reboilers
- Kettle or gravity-fed reboilers

There are dozens of other types of reboilers, but these four represent the majority of applications. Regardless of the type of reboiler used, the following statement is correct: Almost as many towers flood because of reboiler problems as because of tray problems.

The theory of thermosyphon, or natural circulation, can be illustrated by the *airlift pump* shown in Fig. 8.1. This system is being used to recover gold-bearing gravel from the Magdalena River in Colombia, South America. Compressed air is forced to the bottom of the river through the air line. The air is injected into the bottom of the riser tube. The aerated water in the riser tube is less dense than the water in the river. This creates a pressure imbalance between points A and B. Since the pressure at point B is less than that at point A, water (as well as the gold and gravel) is sucked off the bottom of the river and up into the riser tube. We can calculate the pressure difference between points A and B as follows:

$$\frac{(\text{HRW})(\text{DRW}) - (\text{HRT})(\text{DRT})}{2.31} = \Delta P$$

where HRW = height of water above the bottom of the riser, ft
DRW = specific gravity of fluid in the riser; in this case 1.0
HRT = height of the aerated water in the riser tube, ft

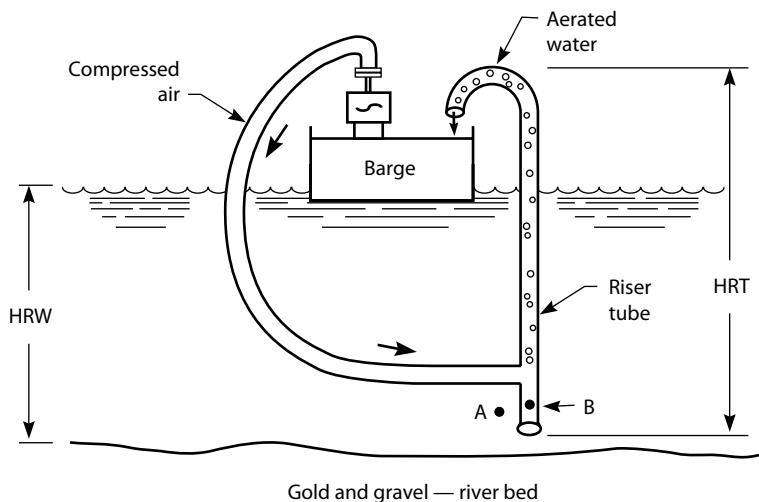


FIGURE 8.1 An airlift pump.

DRT = specific gravity of aerated water in the riser tube
(this number can be obtained only by a trial-and-error calculation procedure)

ΔP = differential pressure between points A and B, psi

In a thermosyphon or natural-circulation reboiler, there is, of course, no source of air. The aerated liquid is a froth or foam produced by the vaporization of the reboiler feed. Without a source of heat, there can be no vaporization. And without vaporization, there will be no circulation. So we can say that the source of energy that drives the circulation in a thermosyphon reboiler is the heating medium to the reboiler.

8.1 Thermosyphon Reboilers

8.1.1 Once-Through Thermosyphon Reboilers

Figure 8.2 shows a once-through thermosyphon reboiler. The driving force to promote flow through this reboiler is the density difference between the reboiler feed line and the froth-filled reboiler return line. For example:

- The specific gravity of the liquid in the reboiler feed line is 0.600.
- The height of liquid above the reboiler inlet is 20 ft.
- The mixed-phase specific gravity of the froth leaving the reboiler is 0.061.

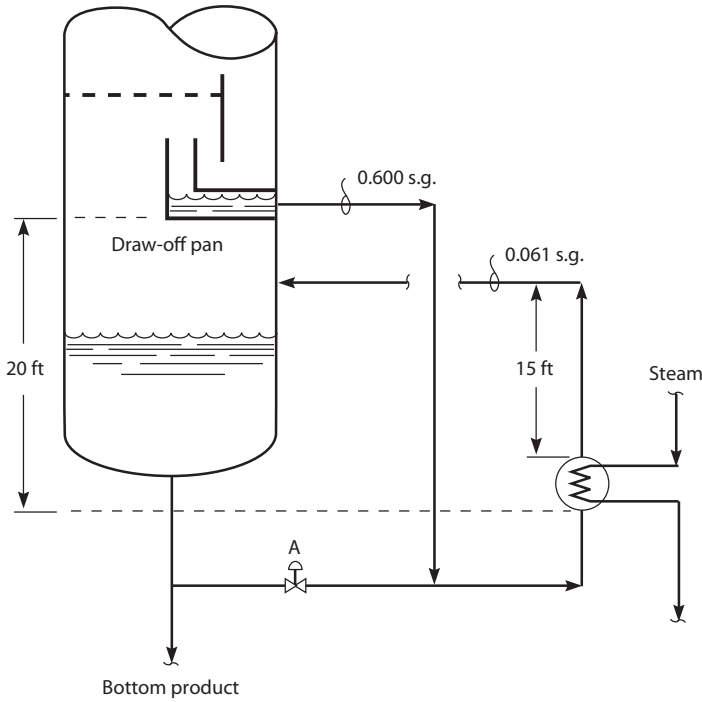


FIGURE 8.2 Once-through thermosyphon reboiler.

- The height of the return line is 15 ft.
- Feet of water per psi = 2.31.

The differential pressure driving force is then

$$\frac{20 \text{ ft} \times 0.600 - 15 \text{ ft} \times 0.061}{2.31} = 4.7 \text{ psi}$$

What happens to this differential pressure of 4.7 psi? It is consumed in overcoming the frictional losses, due to the flow in the

- Reboiler
- Inlet line
- Outlet line
- Nozzles

If these frictional losses are less than the 4.7 psi I have calculated above, then the inlet line does not run liquid full. If the frictional losses are more than the 4.7 psi, the reboiler draw-off pan overflows, and flow to the reboiler is reduced until such time as the frictional losses drop to the available thermosyphon driving force.

The once-through thermosyphon reboiler, shown in Fig. 8.2, operates as follows:

- All the liquid from the bottom tray flows to the reboiler.
- None of the liquid from the bottom of the tower flows to the reboiler.
- All the bottoms product comes from the liquid portion of the reboiler effluent.
- None of the liquid from the bottom tray flows to the bottom of the tower.

This means that when the once-through thermosyphon reboiler is working correctly, the reboiler outlet temperature and the tower-bottom temperature are identical. If the tower-bottom temperature is cooler than the reboiler outlet temperature, something has gone wrong with the thermosyphon circulation.

8.1.1.1 Loss of Once-Through Thermosyphon Circulation

There are several common causes of loss of circulation. The common symptoms of this problem are

- Inability to achieve normal reboiler duty
- Low reflux drum level, accompanied by low tower pressure, even at a low reflux rate
- Bottoms product too light
- Reboiler outlet temperature hotter than the tower-bottom temperature
- Opening the steam or hot-oil inlet heat supply valve does not seem to get more heat into the tower

The typical causes of this problem are

- Bottom tray in tower leaking due to a low dry tray pressure drop
- Bottom tray, seal pan, or draw-off pan is damaged
- Reboiler is partially plugged
- Reboiler feed line is restricted
- Reboiler design pressure drop is excessive
- Tower-bottom liquid level is covering the reboiler vapor return nozzle

If the loss of circulation is due to damage or leakage inside the tower, one can restore flow by opening the start-up line (valve A shown in Fig. 8.2), and raising the liquid level. But if the reboiler is fouled, this will not help.

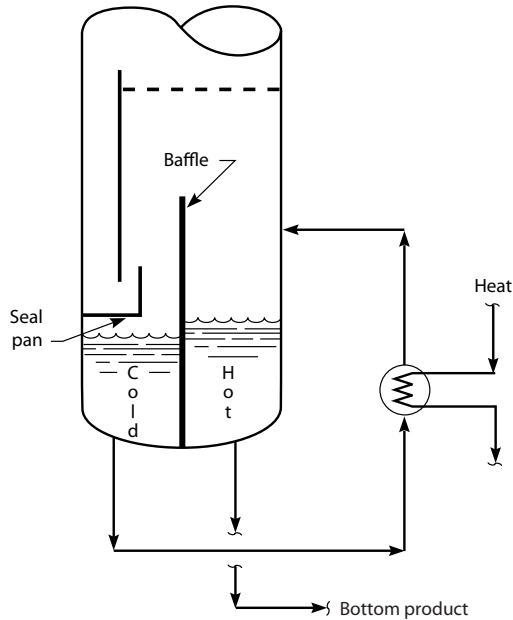


FIGURE 8.3 A once-through reboiler, with a vertical baffle plate.

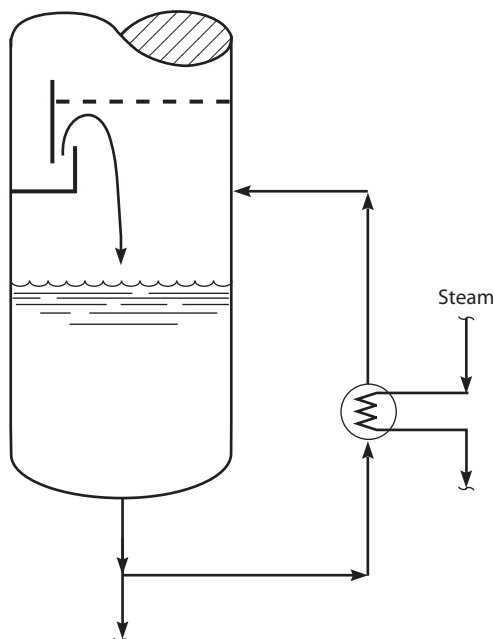
Figure 8.3 shows a once-through thermosyphon reboiler with a *vertical baffle*. This looks quite a bit different from Fig. 8.2, but process-wise, it is the same. Note that the reboiler return liquid flows only to the hot side of the tower bottoms. Putting the reboiler return liquid to the colder side of the tower bottoms represents poor design practice. While most designers do it this way, it is still wrong.

8.1.2 Circulating Thermosyphon Reboilers

My last statement requires some clarification. But to understand my explanation, an understanding of the important differences between a once-through thermosyphon reboiler and a circulating thermosyphon reboiler is critical. Figure 8.4 shows a circulating reboiler. In this reboiler

- The reboiler outlet temperature is always higher than the tower-bottom temperature.
- Some of the liquid from the reboiler outlet will always recirculate back into the reboiler feed.
- Some of the liquid from the bottom tray drops into the bottoms product.
- The tower-bottom product temperature and composition are the same as the temperature and composition of the feed to the reboiler.

FIGURE 8.4 A circulating thermosyphon reboiler.



The liquid feed rate to the once-through thermosyphon reboiler is limited to the amount of liquid overflowing the bottom tray. The liquid feed rate to the circulating thermosyphon reboiler can be quite high—limited only by the available liquid head thermosyphon driving force. However, we should note that the liquid head thermosyphon driving force for a circulating thermosyphon reboiler is proportional to the height of the liquid level in the bottom of the tower above the reboiler inlet nozzle, whereas with a once-through thermosyphon reboiler, as described previously, the corresponding height is the elevation of the floor of the draw-off pan sump above the reboiler inlet nozzle.

For a circulating thermosyphon reboiler, the rate of circulation can be increased by

- Increasing the steam or hot-oil flow through the reboiler. This reduces the specific gravity or density of the froth or foam in the reboiler effluent line.
- Increasing the tower bottoms liquid level. However, should this level reach the reboiler return nozzle, thermosyphon flow will be restricted or even stop. Then the reboiler heat duty will be reduced, and the tower pressure will drop. Sometimes this may cause the tower to flood.

8.1.3 Circulating Versus Once-Through Thermosyphon Reboilers

We said before that it was wrong to return the effluent from a once-through reboiler with a vertical baffle to the cold side of the

tower's bottom. Doing so would actually make the once-through thermosyphon reboiler work more like a circulating reboiler. But if this is bad, then the once-through reboiler must be better than the circulating reboiler. But why?

- The once-through reboiler functions as the bottom theoretical separation stage of the tower. The circulating reboiler does not, because a portion of its effluent back mixes to its feed inlet. This backmixing ruins the separation that can otherwise be achieved in reboilers.
- Regardless of the type of reboiler used, the tower-bottom product temperature has to be the same, so as to meet product specifications. This is shown in Fig. 8.5. However, the reboiler outlet temperature must always be higher in the circulating reboiler than in the once-through reboiler. This means that it is more difficult to transfer heat in the former than in the latter.
- Because the liquid from the bottom tray of a tower with a circulating thermosyphon reboiler is of a composition similar to that of the bottoms product, we can say that the circulating thermosyphon reboiler does not act as a theoretical separation stage. However, the liquid from the bottom tray of a tower with a once-through thermosyphon reboiler can be quite a bit lighter in composition (and hence cooler) than the bottoms product composition, and thus we say that the once-through thermosyphon reboiler does act as a theoretical separation stage. The cooler the liquid flow from the bottom tray of a tower, the less the vapor

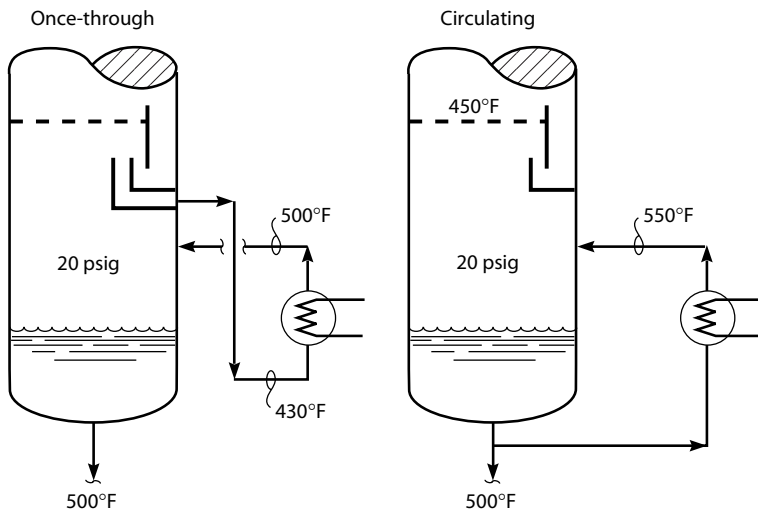


FIGURE 8.5 Once-through vs. circulating reboilers.

flow through that tray. This is because the hot vapor flowing up through a tray heats up the downflowing liquid. This means that there is a greater vapor flow through the bottom tray of a tower with a circulating thermosyphon reboiler than there would be through the bottom tray of a tower with a once-through thermosyphon reboiler. Everything else being equal, then, the tower served by the circulating reboiler is going to flood before the tower served by the once-through reboiler.

8.1.4 The Super-Fractionation Stage

A once-through thermosyphon reboiler not only functions as the bottom theoretical stage of a tower, it also functions as if it is a super-fractionation stage. This means that it is more important than any individual tray in the tower. This is especially true with a multi-component tower bottoms product.

I've devoted a whole chapter to this concept of the super-fractionation (Chap. 49) stage in this book. The concept of once-through versus circulating reboilers and the super-fractionation stage is personally important to me. My perspective of any engineering problem is always the same: How does this problem look to the operator in the field? This is my perspective regardless of whether I am designing new equipment or troubleshooting existing equipment. To the operator, the malfunction of a once-through thermosyphon reboiler appears to be a major malfunction. That is why I've devoted so much space to this subject.

8.1.5 Excessive Thermosyphon Circulation

In a once-through reboiler, the liquid flow coming out of the reboiler is limited to the bottoms product. In a circulating reboiler, the liquid flow coming out of the reboiler can be extremely high. If the reboiler return nozzle is located too close to the bottom tray of the tower, the greater volume of liquid leaving the nozzle can splash against the bottom tray. This alone can cause the entire column to flood. The best way to stop this flooding is to lower the tower bottom level.

Sometimes higher rates of thermosyphon circulation are good. They help prevent fouling and plugging of the reboiler due to low velocity and dirt in the bottoms product and especially high vaporization rates. If the percentage of vaporization in a once-through reboiler is above 60 percent and dirt in the bottoms product is expected, then a circulating reboiler would be the better choice.

8.2 Forced-Circulation Reboilers

Figure 8.6 shows a once-through forced-circulation reboiler. Such a reboiler differs from a thermosyphon reboiler in that it has a pump to *force circulation*, rather than relying on natural or thermosyphon circulation. This extra pump seems rather wasteful—and it is.

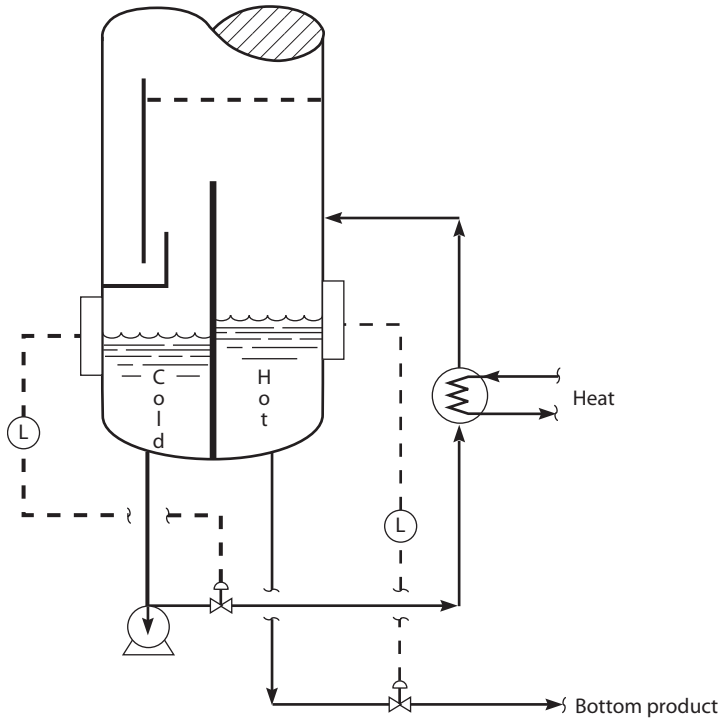


FIGURE 8.6 Forced-circulation once-through reboiler.

The great advantage of forced circulation is that careful calculation of the pressure drop through the reboiler and associated piping is not critical. But as we can see in Fig. 8.6, the operator now has two tower-bottom levels to control. Further, if the hot-side liquid level rises above the reboiler return nozzle, the force of the vapor and liquid rushing back into the column will cause the trays to flood, but the reboiler heat input will not be affected.

Most often, forced circulation is used with fired reboilers. If flow is lost to such a reboiler, furnace tube damage is likely to result. Hopefully, this is less likely to occur with a forced-circulation reboiler. Also, the higher pressure drop of a furnace may force the designer to use a pump. Sometimes we also see a forced-circulation reboiler system if the reboiler heat is to be recovered from a number of dispersed heat sources that are far away from the tower and hence a lot of pressure drop has to be overcome.

8.3 Kettle Reboilers

Reboilers are sometimes inserted into the bottom of a tower. These are called “stab-in” reboilers. It is not a terribly good idea, because

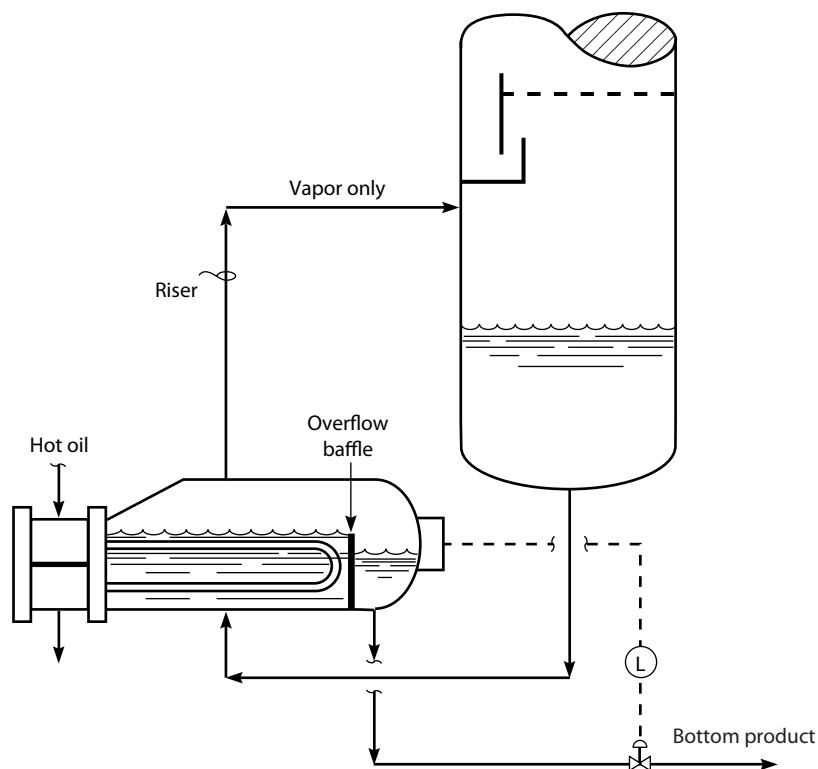


FIGURE 8.7 Kettle reboiler.

it makes it more difficult to fix a leaking or to clean a fouled reboiler without opening the tower itself. However, the “kettle” reboiler, shown in Fig. 8.7, has essentially the same process performance characteristics as the stab-in reboiler, but is entirely external to the tower.

Note that in a kettle reboiler the bottoms product level control valve does not control the level in the tower; it controls the level on the product side of the reboiler only. The liquid level on the boiling or heat-exchanger side of the kettle is controlled by the internal overflow baffle. But what controls the tower-bottom liquid level?

To answer this, let us see how such a gravity-fed or kettle reboiler works:

1. Liquid flows out of the tower into the bottom of the reboiler’s shell.
2. The liquid is partially vaporized.
3. The domed top section of the reboiler separates the vapor and the liquid.

4. The vapor flows back to the tower through the riser line. This is the column's stripping vapor or heat source.
5. The liquid overflows the baffle. The baffle is set high enough to keep the tubes submerged. This liquid is the bottoms product.

The liquid level in the bottom of the tower is the sum of the following factors:

- The nozzle exit loss of the liquid leaving the bottom of the tower
- The liquid feed-line pressure drop
- The shell-side exchanger pressure drop, which includes the effect of the baffle height
- The vapor-line riser pressure drop, including the vapor outlet nozzle loss

Note that it is the elevation, or the static head pressure, in the tower that drives the kettle reboiler. That is why we call it a *gravity-fed reboiler*. Also, the pressure in the kettle will always be higher than the pressure in the tower. This means that an increase in the reboiler heat duty results in an increase of liquid level in the bottom of the tower.

Should the liquid level in the bottom of the tower rise to the reboiler vapor return nozzle, the tower will certainly flood, but the reboiler heat duty will continue. Unfortunately, reboiler shell-side fouling may also lead to tray flooding. This happens because the fouling can cause a pressure-drop buildup on the shell side of the reboiler.

Remember, though, that the increased tower-bottom liquid level will not be reflected on the indicated bottom level seen in the control room, which is actually the level at the end of the kettle reboiler. This is a constant source of confusion to many operators who have towers that flood as a result of high liquid levels, yet their indicated liquid level remains normal.

8.4 Don't Forget Fouling

I have seen towers equipped with kettle reboilers flood due to high liquid levels a dozen times in my career. The story is always the same. The elevation difference between the reboiler vapor return nozzle and the overflow baffle inside the kettle is only 2 or 3 ft. The designer has forgotten about shell-side fouling.

A reasonable allowance for shell-side fouling could be an extra 1 psi of ΔP . If we are dealing with a reboiler feed with a specific gravity

of 0.65 s.g., then the required height of the liquid inside the tower would increase by about 3.5 ft. This would increase the height of the reboiler return nozzle also by 3.5 ft.

In one debutanizer, I had to revamp a new tower by removing the bottom two trays and elevating the reboiler return nozzle by 5 ft. The designer had made no allowance for shell-side fouling. The shut-down of the new debutanizer tower was the price my client paid for the designer's error.

8.5 Vapor Binding in Steam Reboilers

The accumulation of non-condensables on the tube side of horizontal steam reboilers has an extremely detrimental effect on heat transfer. I have observed reboiler duty drop by well over one-half. The origins of the non-condensable vapors are:

- Contamination of the steam supply with CO_2 , N_2 , or light hydrocarbons
- Reboiler tube leaks
- Air trapped inside the channel head on start-up

There are a number of theories as to how these non-condensables reduce the reboiler duty to such a great extent:

- Tubes fill with gas and prevent the flow of steam into those tubes
- The non-condensables produce a film of gas around the tubes
- The non-condensables reduce the steam partial pressure and thus reduce the condensation temperature of the steam

Most of my experience with this subject is a loss in reboiler duty (over one day) due to a tube leak, or the more gradual loss of reboiler duty (over one month) due to CO_2 accumulation in the channel head. Regardless, to alleviate the loss of reboiler duty, vent the channel head just *below* the bottom channel head pass partition baffle.

CHAPTER 9

Inspecting Tower Internals

Tray efficiency is important for meeting product specifications. Even if all products are on spec, we should still be concerned about maximizing tray fractionation efficiency so as to minimize reflux rates. The more efficient the trays, the less reflux we require to produce an acceptable degree of fractionation. Since the reflux originates in the reboiler, reducing reflux also reduces reboiler energy requirements. As energy is our biggest operating cost, tray efficiency is a major component in minimizing operating costs.

The fractionation tray efficiency is largely a function of the vapor-liquid mixing efficiency. The more intimate the contact between the rising vapor and the descending liquid, the better the tray efficiency. The better the tray efficiency, the less fuel we need to run the process.

As I discussed in the previous chapters, tray contacting efficiency is mainly a function of two factors:

- Tray deck levelness
- Weir levelness

9.1 Tray Deck Levelness

For smaller diameter towers a visual check of tray deck levelness is sufficient. For two-pass trays, a small-diameter tower is less than 8 ft. For single-pass trays, a diameter of less than 6 ft is small.

For towers of 10 ft or more in diameter, check for out-of-levelness of a tray check using a carpenter's laser level, available in hardware stores for about \$40. Purchase a level that has short tripod legs. Use the bubble to level up the legs. Set the level on one end of the tower, and check the height of the red beam at the other end and at the center of the tray for out-of-levelness. As it is often dim and dusty in the tower, the trace of the red laser may be clearly visible. Low points and areas of the tray deck that are out of level can now be easily identified.

The more level the tray, the better the mixing efficiency between vapor and the liquid. Certainly, if the tray out-of-levelness is greater than the height of the weir, tray efficiency will be badly degraded.

Checking for weir out-of-levelness is easy. Set the laser level on the edge of the weir. Using the bubble glass level indicator, adjust the laser level to a true horizontal position. The line of red light compared to the top of the weir will indicate how much of the weir is out of level. A weir that is more than 0.5 inch out of level should be re-adjusted. If it is not, stagnant liquid pools behind the higher section of the weir, as described in the prior chapter, will result and ruin the tray's efficiency.

9.2 Loss of Downcomer Seal Due to Leaks

The vertical edges to the downcomers are bolted to bars welded to the vessel wall. These are called "downcomer bolting bars." Gaskets are often used to tightly seal the edge of the tray downcomer to these bars. If the bolts are loose or if the gaskets are missing, vapor will blow into the downcomer and displace the descending liquid. Downcomer backup and flooding may result.

The area underneath the downcomer is called the downpour area. If a tray deck corrodes, it often first holes through in the downpour area. This will cause flooding due to downcomer back-up.

The bottom edge of the downcomer will be somewhat flexible in larger diameter towers. If the width of the tower is less than 5 ft, then the downcomer bolting bars prevent flexing of the bottom edge of the downcomer. However, if the width of the downcomer is over 5 or 6 ft, then downcomer bracing brackets (see Fig. 9.1) are required. The bottom edge of the downcomer should be immobilized by attachment to the bolting bar or bracing bracket every 4 to 5 ft of downcomer width.

Recall that the pressure outside the downcomer is slightly greater than the pressure inside the downcomer. Therefore, a force will push the downcomer toward the vessel wall and reduce the open area of

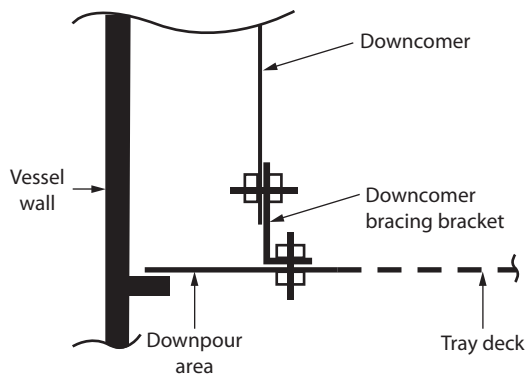


FIGURE 9.1 Downcomer bracing bracket prevents downcomer flexing.

the downcomer. This restriction promotes downcomer backup and flooding. Don't expect to see this deformation of the downcomer during your inspection. Once the vapor flow through the tray stops, the downcomer will spring back to its design position.

9.3 Effect of Missing Caps

I tend to ignore a few missing valve caps, unless many of the caps are missing in a localized area or the percent of missing caps exceeds 5 to 10 percent. The caps serve very little function as the open area underneath the cap is about the same as the hole area covered by the cap. The flow of vapor passing through an orifice without its cap may be about 70 percent greater than the design. Compared to other tray problems, this is not too serious a problem.

9.4 Repairing Loose Tray Panels

Here's a sad story that happened in Corpus Christi, Texas. I inspected the tower and found two problems:

- One or two tray panels were dislodged on each tray.
- The tray panels were encrusted with 0.5 inch of corrosion products.

The plant manager was in a terrible rush to bring the tower back on-line. He instructed the maintenance folks to bolt the trays back up, but skip the lengthy tray cleaning. When the tower was restreamed, the trays flooded at 60 percent of their preturnaround capacity. Fixing the tray panels forced all the vapor to flow through the valve caps. Unfortunately, due to the corrosion deposits on the decks, the tray pressure drop was twice the design ΔP . This raised the liquid levels in the downcomers and thus the flooding. While identifying loose tray panels is fine, it's best to be careful before changing something that's already working.

9.5 Improper Downcomer Clearance

The bottom edge of the downcomer from the tray above should be 0.25 to 0.5 inch below the top edge of the weir of the tray below. This is called a *positive downcomer seal*. Without a positive downcomer seal, vapor will flow up the downcomer and displace the downflowing liquid. This will cause flooding due to excessive downcomer backup.

On the other hand, if the bottom of the downcomer is too close to the tray below, then the "head loss under the downcomer" will be excessive. Typically, a minimum downcomer clearance is 1.5 to 2 inch. Too small a downcomer clearance will result in restricting the liquid flow from the downcomer. This will also cause excessive downcomer backup and flooding. Check the correct downcomer clearance on the vendor tray drawings prior to the tower inspection.

9.6 Inlet Weirs

Most trays have outlet weirs devoted to maintaining the downcomer seal. But some trays have inlet weirs too, or inlet weirs, but no outlet weirs. A sketch of an inlet weir is shown in Fig. 9.2. Note the horizontal distance between the downcomer and the inlet weir (dimension x). This distance ought to be equal to or greater than the downcomer clearance—that is, the vertical space between the tray floor and the bottom edge of the downcomer. Unfortunately, a small deformation of the downcomer may push the downcomer quite close to the inlet weir. The resulting reduction in the horizontal clearance between the inlet weir and the downcomer will restrict the liquid flow. This will cause downcomer backup and tray flooding of the trays above.

Often, there is no process reason for the use of inlet weirs, especially at higher liquid rates. Then the inlet weirs may be removed. But some tray types, such as “Exxon Jet Tab” trays or total trap-out chimney trays with no outlet weir, absolutely require the use of inlet weirs.

9.7 Seal Pans

The downcomer from the bottom tray is submerged in a seal pan (see Fig. 9.3) to preserve its downcomer seal. I always set the horizontal dimension between the over-flow lip of the seal pan (dimension y) and the downcomer at 4 inches, so I never have to worry about restricting liquid flow from the bottom tray. This horizontal dimension should be equal to or greater than the vertical clearance between the downcomer and the seal pan floor (dimension x , which is typically 2 to 3 inches). If a deformation of the downcomer reduces the horizontal clearance between the seal pan overflow lip and the downcomer,

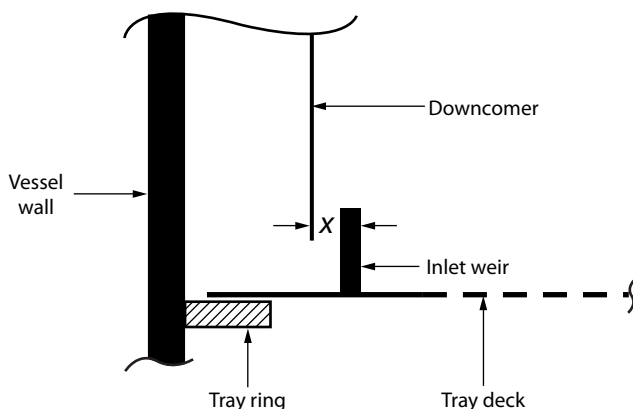


FIGURE 9.2 Inlet weirs can cause flooding.

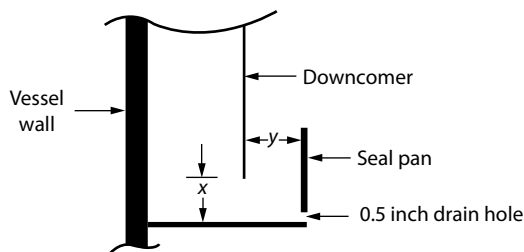


FIGURE 9.3 Reduction of seal pan to downcomer clearance can flood tower.

the resulting restriction can cause the bottom tray to flood due to downcomer back-up. If the bottom tray floods, flooding will progress up the column. With time, the entire column will flood due to the small restriction in the seal pan. That's why a detailed tray-by-tray inspection is important.

9.8 Drain Holes

All tray sections, draw-off sumps, recessed downcomer inlet sumps, seal pans, chimney trays, and other tower internals must be self-draining. As a column is heated on start-up, water can expand to steam with explosive force. Especially for lower pressure and larger diameter towers, tray panels are frequently dislodged due to flashing of trapped water. My experience indicates that drain holes of less than 0.5 inch, drilled in the bottom of a pan are likely to plug off due to fouling. The better way to place a drain hole is in a vertical wall of a pan, flush with the pan floor, as indicated in Fig. 9.3.

Chimney draw-off trays are typically designed to slope toward the draw-off nozzle, thus avoiding a need for drain holes. However, with age, such trays warp because of thermal expansion and pools of water can be trapped. Inspection after water washing will reveal such poorly drained trays and pinpoint the location of required drain holes. But excessive use of drain holes can also be bad. Product losses through the drain holes can be estimated by the following:

$$H = \frac{V^2}{2}$$

where H = liquid level on tray, inches

V = hole velocity, feet per second

The volumetric rate of flow lost through the drain hole is then the hole area multiplied by the hole velocity. With this equation in hand, your inspection may find drain holes that are best closed.

If a drain hole plugs on a bottom tray seal pan, I have found that the seal pan will, with time, fill with dirt and corrosion products. This restricts the flow from the downcomer and has caused at least one amine fuel gas hydrogen sulfide absorber to flood. To avoid this problem, I'll size the drain holes in such seal pans to pass 20 percent of the liquid flow, or at least use a single 1-inch drain hole. Unless liquid is flushed out of the bottom of the bottom tray seal pan, the solids washed down the column settle out in the seal pan.

9.9 Vortex Breakers

I have a personal dislike of this widely used device. The idea of the vortex breaker is to prevent swirling liquid from conveying vapor into the draw-off nozzle. Sometimes though, trash lost in the tower during a turnaround gets caught on the vortex breaker. Since the draw nozzle typically is located in a sump below the seal pan, it may be hard to inspect. In some cases, it would be best to just cut off the vortex breaker. This is fine if the nozzle exit velocity is less than 3 or 4 ft/s.

The seal pan itself is also hard to inspect, as it is usually hidden inside the draw-off sump. Corrosion deposits may accumulate in the seal pan and cause the trays above to flood. A small mirror is a good tool to observe these hard to reach locations: maintenance folks use this tool all the time.

9.10 Chimney Tray Leakage

I like to establish a water level on chimney trays up to the overflow pipe. Next, I crawl under the tray to look for trickles of water. When the tower heats up, such minor leaks are likely to get worse. Also, I check the alignment of the chimney hats to make sure they are actually centered over the chimneys. After the water is drained down, I look for puddles of water that indicate a need for drain holes. If there is an overflow pipe, I'll check that the overflow pipe will be submerged in its seal pan.

9.11 Shear Clips

If a tower has a history of tray deck damage due to pressure surge or high liquid level, the mechanical integrity of the trays should be upgraded. This is done by the use of shear clips, as shown in Fig. 9.4. The use of shear clips is not the best way to improve the mechanical integrity of trays but it is the only effective method to use during a turnaround, when no other plans have been made to mechanically upgrade the tray decks and time is short.

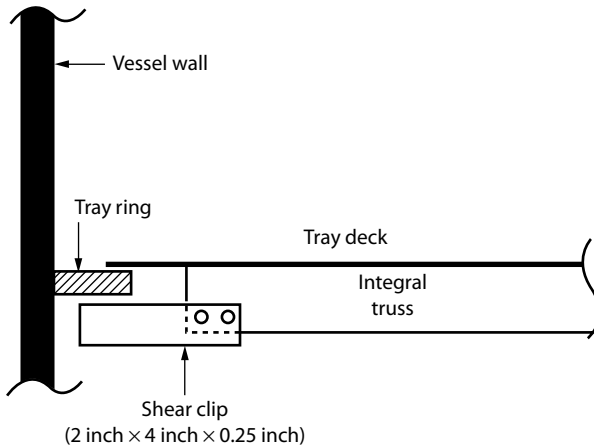


FIGURE 9.4 Shear clips improve the mechanical integrity of the tray deck.

Underneath the tray, where the tray panels are bolted together, there is a narrow vertical strip of steel, which is called the integral truss. This truss is not connected to the tray ring. Referring to Fig. 9.4, a small steel bar (4 inch by 2 inch by 0.25 inch) is welded or bolted to the end of the integral truss. This bar is the shear clip, which is inserted underneath the tray ring. (Do not weld the shear clip to the tray ring.) When an upward surge of vapor pushes the tray up, the force is transmitted along the length of the integral truss, through the shear clip, to the tray ring, and thus to the vessel wall. Many large diameter towers will already have shear clips. But if your inspection indicates they are not present and tray failure has been a problem, the installation of shear clips is the way to go for three reasons:

- The job can usually be done in 24 hours, while other tower work continues.
- The shear clips can be cut from an ordinary 0.25-inch carbon steel plate.
- Experience proves they are effective in resisting moderate pressure surges.

9.12 Bubble-Cap Trays

About 5 percent of the trayed towers have old style bubble-cap trays. Dirt and coke tends to accumulate under the cap. This cannot be seen unless the cap is removed. Coke underneath the bubble cap will restrict the vapor flow, increase vapor ΔP and promote downcomer

backup and flooding. It only takes a moment to unscrew a bubble cap using a crescent wrench. The maintenance folks will usually clean the deck, but not remove the caps. Liz injured herself crawling through a manway helping me discover this problem at the Motiva Refinery visbreaker fractionator in Convent, Louisiana. This tower flooded on start-up, because the small areas between the caps and the chimneys inside the caps were mostly coked off.

9.13 Final Inspection

At a Gulf Coast (Amoco, Texas City) refinery, the reboiler thermosyphon circulation could not be re-established after a turnaround. The tower was re-opened and a less-than-alive contract employee was found stuck in the reboiler draw-off nozzle. At the Good Hope Refinery (when I was the technical manager), we once left a complete scaffold (poles, boards, everything) in the bottom of a debutanizer tower. Rags, hard hats, plywood, and especially plastic bags left in packed columns should be removed from inside draw sumps and downcomers. I know it's rough on the knees, but crawl across every tray and look into each downcomer. One lost flashlight in a small downcomer may flood every tray in the tower. A rag caught on a vortex breaker in a jet fuel draw box has caused a complete refinery shutdown.

Check the tray clips, tray panels, and downcomer bolting bars. At the least the nuts and bolts should be finger-tight. If you find a single loose nut, insist that every nut on that tray be retightened. I will check the tray clips and 10 percent of the downcomer bolting bar nuts for finger-tight.

9.13.1 Tower Internal Manways

Contract maintenance workers often will not replace the tray manways unless the tray manway is adjacent to a tower external manway. They reason that once the tray manways that are visible from the tower manway are closed, there is no way for someone to inspect the other trays. This problem is not just common—it is universal. The maintenance force at the Good Hope Refinery pulled this nasty trick on me at the coker fractionator. Equipped with my crescent wrench, I opened the tray internal manway below the side tower manway. I discovered that the 12 trays below this point had their manways stacked in their downcomers. In 1990, I worked on a project to improve fractionation at the Chevron Refinery crude distillation unit in El Segundo, California. When the tower was opened to implement my design, the tray manways were found lying on the tray decks below the diesel draw tray. The lesson is, inspect each tray and then witness the closure of each tray manway, separately.

9.14 Conclusion

The most important job of the process engineer working in a refinery or chemical plant is to inspect tower internals during a turnaround. The unit engineer and the panel board operator should go through the tower together. Inspecting a 40-tray, two-pass crude distillation tower properly will take me 3 to 4 hours. It's a dirty, difficult job. I once inspected such a tower at the Coffeyville, Kansas, refinery (remember the Dalton Brothers bank robbery in Coffeyville). What has helped me at this work is the experience of growing up in New York. I learned then that it's best not to trust anyone. It's best to see it yourself. Also, crawling through the trays is the only real way to understand how they work and what malfunctions can be expected.

I find it best to make a list of all nozzle locations relative to the tray number. This helps me check my location inside the tower. Also, I make a list of dimensions from the vendor tray drawings for easy reference in the column:

- Weir height
- Downcomer width and clearance
- Seal and draw-off pans

Most of my clients now use a digital camera to record tray conditions. For me, this is good for record keeping only, not for decision making as to what tray repairs are needed. I suggest you make a copy of the checklist below, and refer to it when doing your next tower internal inspection.

9.14.1 Tower Checklist Items

1. Downcomer sealed below weirs.
2. Weirs level.
3. Bottom of downcomers level.
4. Tray clips tight. Trays bolted together tight.
5. Downcomer bolting bars tight.
6. Downcomer bracing brackets installed.
7. No holes in downpour plates (i.e., area below downcomers).
8. Tray decks level.
9. All sections self-drainable.
10. Seal pans intact and drainable.
11. Draw-off sumps intact and drainable.
12. Distributors not rigidly supported.
13. Distributor flanges double-bolted tight and gaskets installed.

14. Draw-off nozzles clear.
15. Explosion doors open and close freely.
16. Tray deck panels aligned correctly (as per design).
17. Tray deck manways reinstalled.
18. Tower cleared of flashlights, rags, gloves, plastic bags, wrenches, and hard hats.

Reference

1. N. P. Lieberman, *Troubleshooting Process Operations*, 4th ed. (PennWell).

CHAPTER 10

How Instruments Work

Levels, Pressures, Flows, and Temperatures

In this age of advanced computer control, problems are still caused by the field instruments. The field instruments we discuss in this chapter are

- Level indicators
- Pressure indicators
- Flow indicators
- Temperature indicators

This chapter is particularly important when we consider that the data displayed in the control room are for operator control of the process. Data for engineering purposes should be obtained locally, at the instrument itself. Further, a large percent of control problems are actually process malfunction problems.

10.1 Level

10.1.1 Level Indication

What is the difference between a *gauge glass* and a *level glass*? Simple! There is no such thing as a level glass. The liquid level shown in a gauge glass does not correspond to the level in a process vessel. Figure 10.1 is a good example. This is the bottom of an amine fuel-gas

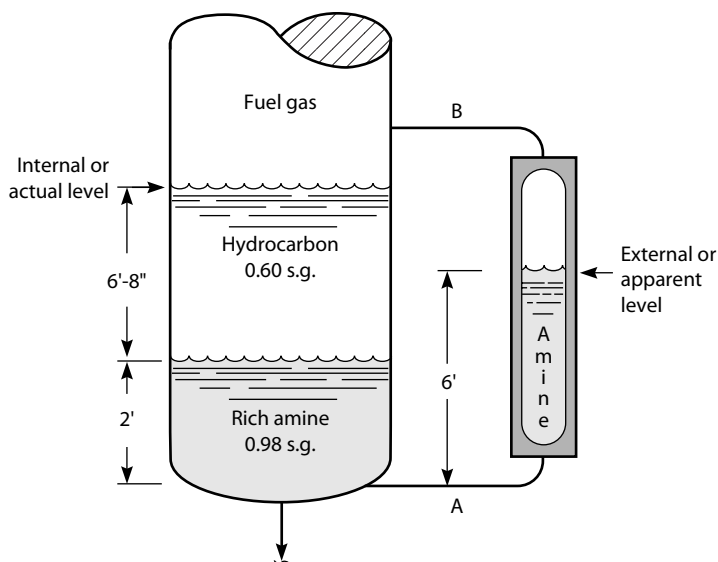


FIGURE 10.1 A gauge glass functions as a manometer.

absorber. This tower is used to remove hydrogen sulfide from the fuel gas. At the bottom of the tower, there are three phases:

- Fuel gas: 0.01 specific gravity
- Hydrocarbon liquid: 0.60 specific gravity
- Rich amine: 0.98 specific gravity

Because of the location of the level taps of the gauge glass, only the amine is in the glass. The gauge glass simply measures the pressure difference between two points of the tower (points A and B in Fig. 10.1). That is, the gauge glass functions as a manometer, which measures the pressure difference in terms of the specific gravity of the liquid in the gauge glass. Should the specific gravity of the liquid in the glass be the same as that of the liquid in the tower, both the gauge-glass level and the tower level would be the same. But this is never so. The specific gravity of the liquid in the gauge glass is always greater than the specific gravity of the liquid in the tower. Hence, the apparent liquid level in the gauge glass is always somewhat lower than the actual liquid level in the tower.

This discrepancy between the *apparent level* in the gauge glass and the *actual level* (see Fig. 10.1) in the tower also occurs in any other type of level-measuring device. This includes external float chambers, “kidneys,” displacement chambers, and level-trols. The one exception to this is level-measuring devices using radiation techniques.

The three causes of the discrepancy between the external level and the internal level are

- Foam formation inside tower
- Ambient-heat loss from the external gauge glass or level-trol
- The liquid specific gravity in the glass being greater than the specific gravity in the tower, as shown in Fig. 10.1

10.1.2 Level Discrepancies

Let's assume that the gauge glass shown in Fig. 10.1 holds 6 ft of amine. Since the bottom tap is in the amine phase and the top tap is in the gas phase, the liquid hydrocarbon is excluded from the gauge glass. To balance out the weight of the 6 ft of amine, the tower would have to have about 2 ft of amine and 6 ft 8 inch of liquid hydrocarbon. That is, the tower liquid level would be about 8 ft 8 inch or 2 ft 8 inch higher than the gauge-glass level.

If you conclude from the above that we could use the gauge-glass level to actually calculate the level inside the tower, you are quite wrong. To perform the preceding calculation, one would have to assume the *ratio of the phases*. But this is an assumption equivalent to assuming the answer. How, then, does one determine the actual liquid level in the tower on the basis of the apparent liquid level in the gauge glass? The answer is that there is no answer. It cannot be done! And this statement applies to all other sorts of level-measuring instruments—with the exception of radiation devices.

10.1.3 Effects of Temperature on Level

The gauge glass will normally be somewhat colder than the process vessel as a result of ambient-heat losses (an exception to this would be a refrigerated process). For every 100°F decrease in the gauge-glass temperature or level-trol temperature, the specific gravity of the liquid in the glass increases by 5 percent. This rule of thumb is typical for hydrocarbons only. Aqueous (water-based) fluids are totally different.

For example, suppose the height of hydrocarbon liquid in a gauge glass is 4 ft between the level taps. The glass temperature is 60°F. The tower temperature is 560°F. How much higher is the height of liquid in the tower than in the glass? (Answer: 1 ft.)

10.1.3.1 Explanation

$$500^{\circ}\text{F} \times \frac{5}{100} \times \frac{1}{100^{\circ}\text{F}} = \frac{25}{100}$$

- This means that the liquid in the gauge glass is 25 percent more dense than the liquid in the tower bottom.

- Assuming a linear relationship between density and volume, the level of liquid in the tower above the bottom tap of the gauge glass must be

$$\left(1 + \frac{25}{100}\right) \times 4 \text{ ft} = 5 \text{ ft}$$

- In other words, the liquid in the tower is 1 ft above the level shown in the glass.

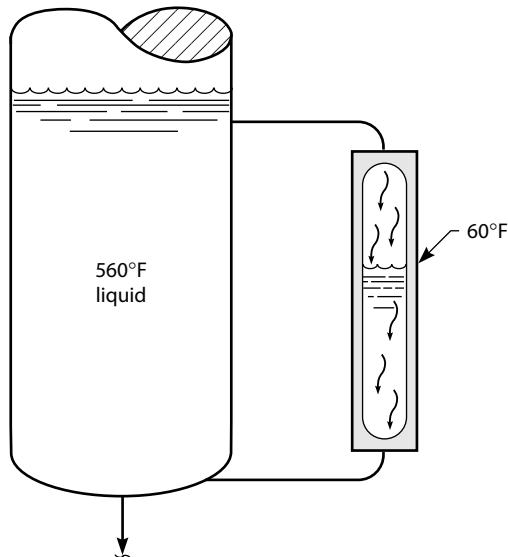
10.1.4 Plugged Taps

How do plugged level-sensing taps affect the apparent liquid level in a vessel? Let's assume that the vapor in the vessel could be fully condensed at the temperature in the gauge glass. If the bottom tap is closed, the level will go up because the condensing vapors cannot drain out of the glass. If the top tap is closed, the level will go up because the condensing vapors create an area of low pressure, which draws the liquid up the glass through the bottom tap. Thus, if either the top or bottom taps plug, the result is a false high-level indication.

10.1.5 High Liquid Level

In our calculation above, we had 4 ft of liquid in the glass and 5 ft of liquid in the tower. But what happens if the distance between the two taps is 4 ft 6 inch? I have drawn a picture of the observed result in Fig. 10.2. Liquid circulates through the glass, pouring through the top tap, and draining through the bottom tap. The apparent liquid level

FIGURE 10.2 Liquid circulation through a gauge glass.



would then be somewhere between 4 ft 0 inch and 4 ft 6 inch; let's say 4 ft 2 inch. The indicated liquid level on the control room chart would then be 92 percent (i.e., 4 ft 2 inch \div 4 ft 6 inch). As the liquid level in the tower increases from 5 ft to 1000 ft, the indicated liquid level would remain at 92 percent.

Once the actual liquid level inside the tower bottom rises above the top-level tap, no further increase in level can be observed in the gauge glass. We say the level indication is "tapped out."

The same sort of problem arises in a *level-trol*, which measures and transmits a process vessel liquid level to the control center. As shown in Fig. 10.3, the level-trol operates by means of two *pressure transducers*, devices for converting a pressure signal into a small electric current. The difference between the two pressure transducers shown in Fig. 10.3 is called the *milliamp* (milliampere) *output*. This milliamp output is proportional to the pressure difference between the bottom and top taps in the level-trol. To convert the milliamp output signal from the level-trol into a level indication, the instrument technician must assume a specific gravity:

$$\text{Percent level} \sim \frac{\text{milliamp signal}}{\text{specific gravity}}$$

But which specific gravity should the instrument technician select? The specific gravity of the liquid in the level-trol, or the lower specific gravity of the liquid in the hotter process vessel? The technician should use the specific gravity in the process vessel and ignore

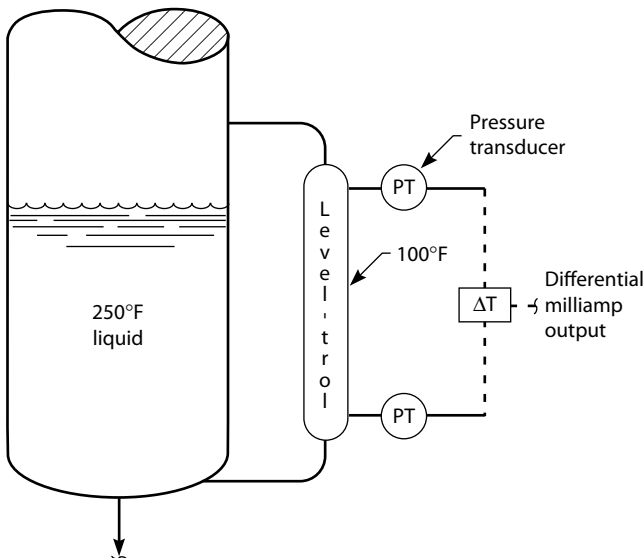


FIGURE 10.3 Operation of a level-trol.

the specific gravity of the liquid in the level-trol. This can be especially confusing, if the operator then compares the apparently low liquid level in the gauge glass to the indicated higher liquid level on the control panel.

10.2 Foam Affects Levels

Trying to predict a vessel level based on the output from the pressure transducers would work only if one knew the actual specific gravity of the fluid in the bottom of a distillation tower. But anyone who has ever poured out a glass of beer realizes that this is not possible. For one thing, the ratio of white froth to yellow beer is never known in advance. Also, the density of the froth itself is unknown and is quite variable.

Figure 10.4 shows a distillation tower served by a circulating thermosyphon reboiler. To some unknown extent, some foam will always be found in the bottom of such vessels. Not sometimes, but always. Why?

The purpose of a tray is to mix vapor and liquid. This produces aerated liquid—or foam. The purpose of a reboiler is to produce vapor. In a circulating reboiler, the reboiler effluent flows up the riser as a froth. Of course, the flow from the bottom of the tower is going to be a clear liquid. Foam cannot be pumped. But there will always be some ratio of foam to clear liquid in the bottom of the tower, and we have no method of determining this ratio or even the density of the foam.

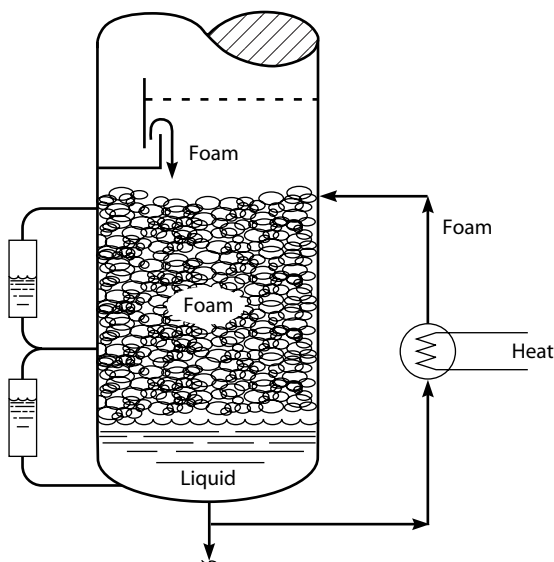


FIGURE 10.4 Split liquid-level indication caused by foam.

Well, if we do not know the average specific gravity of the foamy liquid in the bottom of a tower, how can we find the level of foam in the tower? The unfortunate answer is that, short of using radiation techniques, we cannot.

10.2.1 Split Liquid Levels

The two gauge glasses shown in Fig. 10.4 both show a liquid level. Many of you may have observed this on a process vessel. We certainly cannot have layers of liquid-vapor-liquid-vapor in the vessel. Rather, these split liquid levels are a positive indication of foam or froth in the bottom of the tower.

If the foam is spanning both taps on a gauge glass, then the height of the liquid in the glass is a measure of the specific gravity or density of the foam in terms of the specific gravity of the liquid in the glass. If the foam is above the top tap of both the gauge glasses in Fig. 10.4, then there will be a level in both glasses. The upper gauge glass will show a lower level because the light foam in the tower floats on the top of the heavier foam. Note that these split liquid levels, so often seen in a process vessel, tell us nothing about the real liquid level in the vessel. They are a sign of foam.

Figure 10.5 is a plot of the liquid level in a crude preflash drum versus time. We were steadily withdrawing 10 percent more flashed crude from the bottoms pump than the inlet crude feed rate. The rate of decline of the liquid level noted in the control center was only about 25 percent of our calculated rate. Suddenly, when the apparent level in the control room had reached 40 percent, the level indication started to decline much more rapidly. Why?

This extreme non-linear response of a level to a step change in a flow rate is quite common. Before the sudden decline in the indicated

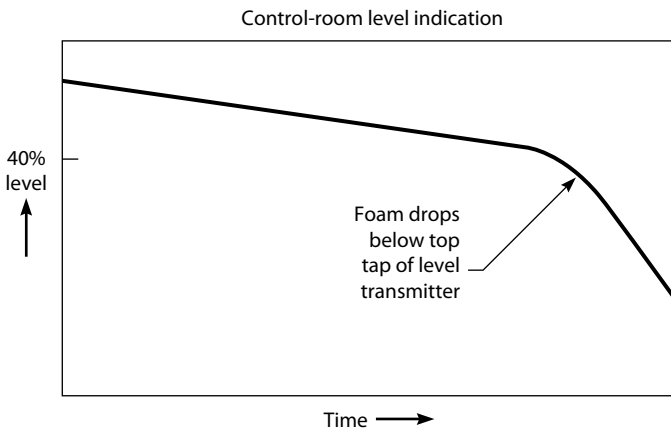


FIGURE 10.5 Foam creates a non-linear response in level indication.

liquid level, foam had filled the drum above the top-level tap. The initial slow decline in the apparent level was due to a dense foam dropping between the level taps being replaced by a lighter foam. Only when the foam level actually dropped below the top tap of the drum did the indicated liquid level begin to decline at a rate representing the actual decline in the level. Thus, we can see that this common, non-linear response is not due to instrument malfunctions, but is a sure sign of foam or froth.

10.2.2 Radiation Level Detection

The only way around the sort of problems discussed above is to use neutrons or x-rays to measure the density in a vessel. In a modern petroleum refinery, perhaps 5 percent of levels are monitored with radiation. It is both safe and effective.

The neutron backscatter technique is best performed with hydrogen-containing products. Both the source of the slow neutrons and the receiver are located in the same box. The slow neutrons bounce off protons (hydrogen ions) and are reflected back. The rate at which these neutrons are reflected back is measured, and corresponds to the hydrocarbon density in the vessel. This measurement is not affected by steel components inside—or outside—the vessel.

X-ray level detection works with a source of radiation and a receiver, located on either side of the vessel. As the percent absorption of the radiation increases, the receiver sees fewer x-rays and a higher density is implied. The x-rays are absorbed by steel components, such as ladders and manways, which can sometimes be confusing.

Either method discriminates nicely between clear liquid, foam, or vapor. Such a level controller can be calibrated to hold a foam level or a liquid level. Of course, this sort of level detection is far more expensive than conventional techniques.

10.2.3 Lost Radiation Source

But let me be completely honest. I once had a radiation level detector on a spent sulfuric acid tank. I was the operating superintendent at this plant, which regenerated sulfuric acid. We had a not-so-small fire at this tank. Rather than repair the tank, pumps, and lines, I decided to dismantle and haul away the entire mess of steel. After all, I had much larger and more modern storage tanks at my disposal.

After the dismantled equipment had been trucked away, my boss said, “And where, Lieberman, is the nuclear level detector radiation source? I imagine even you must realize that these radioactive sources are licensed. God help us if you have lost it.”

I explained to him that man does not live by reason alone. That my loss of the radiation source for the level detector would not affect the ultimate fate of the universe. Unfortunately, my boss had a narrow-minded view of this incident.

10.3 Pressure

10.3.1 Pressure Indicators

The chief engineer of a West Texas process plant had decided to replace the main condenser. Colder weather always coincided with a vastly improved vacuum in their vacuum tower. It seemed as if colder air to the condenser really helped. So the chief engineer concluded that a bigger condenser would also help during warm weather.

Wrong! The chief engineer failed to realize that the vacuum pressure indicator was not equipped with a *barometric pressure compensator*. An ordinary vacuum pressure indicator or pressure gauge reads the pressure difference between the vacuum system and atmospheric pressure. When ambient temperatures drop, the barometer rises or ambient pressure goes up. An ordinary vacuum pressure gauge or indicator would then read an improved vacuum. But in reality, the vacuum has not changed.

The opposite problem would occur in Denver—the Mile-High City. At sea level, full vacuum is 30 inches of mercury (or 30 inches Hg). But in Denver, full vacuum is about 26 inches Hg. An ordinary vacuum pressure gauge reads zero inches of mercury (0 inch Hg) in Denver and in New Orleans, because although these cities are at different altitudes, the vacuum pressure gauge compares system pressure only with ambient pressure. But a vacuum pressure gauge reading 25 inches Hg in New Orleans would correspond to a poor vacuum of 5 inches Hg absolute pressure (30 inches Hg–25 inches Hg). A vacuum pressure gauge reading 25 inches Hg in Denver would correspond to an excellent vacuum of 1 inch Hg absolute pressure (26 inches Hg–25 inches Hg).

All these complications can be avoided when making field measurements by using the vacuum manometer shown in Fig. 10.6.

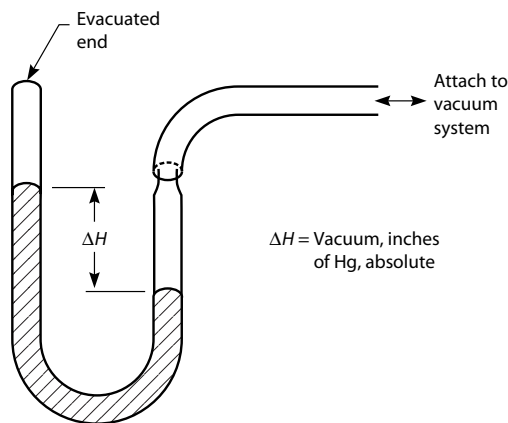


FIGURE 10.6 A mercury absolute-pressure manometer.

The difference between the two mercury levels is the correct inches of mercury absolute pressure, or millimeters of mercury (mm Hg). A drop of water on the evacuated end of the manometer will result in a falsely low vacuum reading.

10.3.2 Pressure Transducers

Disassemble a pressure transducer, and you will see a small plastic diaphragm. A change in pressure distorts this diaphragm and generates a small electrical signal. The signal must be quite tiny, because placing your hand on the transducer can alter its reading. A modern digital pressure gauge uses a pressure transducer. This type of gauge, if zeroed at sea level in New Orleans, will read 4 inches of Hg vacuum in Denver. Most pressure signals transmitted from the field into the control center are generated from pressure transducers. Differential pressure indicators simply take the differential readings from two transducers and generate a milliamp output signal.

10.3.3 Pressure-Point Location

Locating a pressure tap in an area of high velocity is likely to produce a lower pressure indication than the real flowing pressure. Using a purge gas to keep a pressure tap from plugging often can cause a high pressure reading if too much purge gas or steam is used. A pressure tap located below a liquid level will read too high; pressures should be measured in the vapor phase. A pressure tap opposite an inlet nozzle with a high velocity will read higher than the real pressure.

10.4 Flow

10.4.1 Flow Indication

The standard method of measuring flows in a process plant is by use of the orifice plate and orifice flanges, shown in Fig. 10.7. Actually, we rarely measure flows directly. More commonly, we measure the pressure drop across an orifice plate. This pressure drop is due to the increase in kinetic energy of a fluid as it accelerates through the small hole in the orifice plate. The energy to provide the increased velocity comes from the pressure of the flowing fluid in accordance with the following:

$$\Delta P = K \frac{D_f}{62.3} V^2$$

where ΔP = measured pressure drop through the orifice plate, in inches of water (multiply the measured pressure drop in psi by 2.77 to obtain the inches of water ΔP)

V = velocity of the fluid through the orifice plate, ft/s

D_f = density of the fluid, whether vapor or liquid, lb/ft³

K = an orifice coefficient

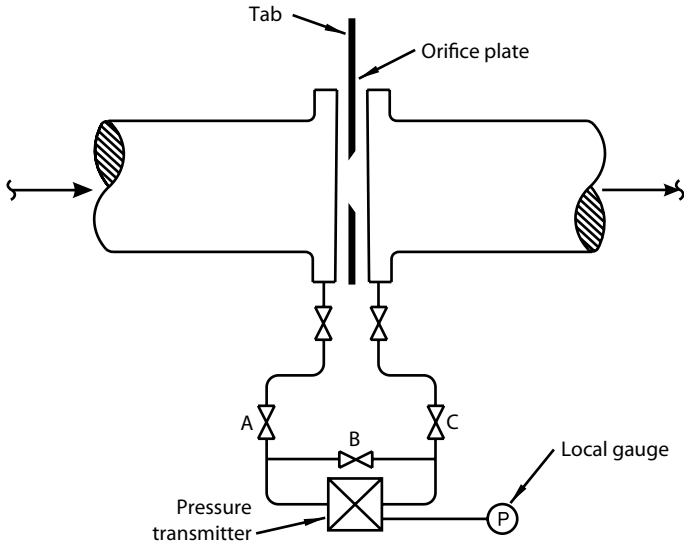


FIGURE 10.7 Flowmeter assembly.

You should look up the orifice coefficient K in your Cameron or Crane handbook—but it is typically a number like 0.6 to 0.8.

10.4.2 Checking Flows in the Field

The competent engineer does not assume a flow indication shown on the control panel is correct, but proceeds as follows:

1. Referring to Fig. 10.7, place an easy-to-read pressure gauge in the position shown. I like to use a digital gauge.
2. By opening both valves A and B, with C closed, you will now be reading the upstream pressure.
3. By opening valve C, with A and B closed, you will read the pressure downstream of the orifice plate.
4. The difference between the two readings is ΔP in the preceding equation. Now solve for V .
5. Look at the tab sticking out of the orifice flanges (see Fig. 10.7). If the orifice plate is installed in the correct direction, there will be a number stamped onto the tab, toward the flow. This is the orifice plate hole diameter; for example, if you see 0.374" stamped on the tab, the orifice hole diameter should be 0.374 inch.
6. Using the hole diameter, calculate the volume of fluid.

Suppose the flowing fluid is water. Assume the area of the orifice is 0.1 ft^2 . The observed $\Delta P = 4 \text{ psi}$. Converting psi to inches of water:

$$1 \text{ psi} = 2.77 \text{ inches H}_2\text{O}$$

Therefore the observed ΔP expressed as inches H_2O is:

$$4 \times 27.7 = 110.8 \text{ inches } H_2O$$

The linear flow rate (V) in ft/s is found from

$$\Delta P \text{ inches } H_2O = \text{orifice coefficient} \times V^2$$

Suppose the orifice coefficient is 0.6 (which would be typical); then:

$$110.8 = 0.6 \times V^2$$

$$V^2 = \frac{110.8}{0.6}$$

$$V^2 = 184.66$$

$$V = \sqrt{184.66}$$

$$V = 13.6$$

We find the volumetric flow rate (F) in ft^3/s after multiplying by the cross-sectional area of the orifice opening, thus in this case:

$$F = 13.6 \text{ ft/s} \times 0.1 \text{ ft}^2 = 1.36 \text{ ft}^3/\text{s}$$

You may notice when you measure ΔP that it is a small value, difficult to measure accurately. This means that the orifice plate hole is oversized, and that the accuracy of the recorded flow on the control panel is also poor, or the measured ΔP is quite high. This means that a lot of pressure is being wasted, and the orifice plate hole is undersized and restricting flow. Furthermore, the recorded flow on the control panel may be off scale.

The reason the orifice flanges are kept close to the orifice plate is that when the liquid velocity decreases downstream of the orifice plate, the pressure of the liquid goes partly back up. Figure 10.8

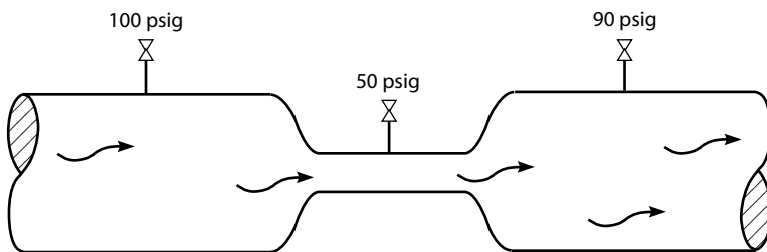


FIGURE 10.8 Pressure recovery for water flow in pipe.

illustrates this point. It is called *pressure recovery*. Whenever the velocity of a flowing fluid (vapor or liquid) decreases, its pressure goes partly back up. An extreme example of this is *water hammer*. The reason the pressure at the end of the pipe in Fig. 10.8 is lower than at the inlet to the pipe is due to frictional losses.

The orifice coefficient K takes into account both frictional pressure losses, and conversion of pressure to velocity. The frictional losses represent an *irreversible process*. The conversion of pressure to velocity represent a *reversible process*.

10.4.3 Other Flow-Measuring Methods

A better way of measuring flows than the ordinary orifice plate method is by inducing vortex shedding across a tube in the flowing liquid and then measuring the velocity of the vortices. This is a nice method, as there are no orifice taps to plug. Then there are Doppler meters, which measure the velocity of a fluid based on how the speed of sound is affected by the flow in a pipe. More commonly, we have rotometers, which measure how far a ball or float is lifted in a vertical tube by the velocity of the liquid. But regardless of their relative merits, perhaps simply for historical reasons, the vast majority of flows in most process plants are measured with the orifice plate flowmeter, shown in Fig. 10.7.

10.4.4 Glycol-Filled Instrument Lines

Many of us have seen the following tag attached to level-sensing lines or to a flow transmitter: “Do not drain—glycol-filled.” This means the instrument mechanic has filled the lines with glycol, mainly for winter freeze protection. Many process streams contain water, which can settle out at low points and, in effect, plug the impulse lines to flow- or level-sensing ΔP transmitters when water freezes. Note that there is not a lot of difference between measuring a flow and a level; they both are typically measured by using a differential pressure signal.

Naturally, just like level indicators, the flow orifice taps can plug. If the upstream tap plugs, the flow will read low or zero. It is best to blow the tap back with glycol, but that is not always practical. If you blow the taps out with the pressure of the process stream, you do not need to refill the impulse lines with glycol to get a correct flow reading. But the lines have to be totally refilled with the same fluid. If you are measuring the flow of a single-phase liquid, just open valves A, B, and C (shown in Fig. 10.7) for a few minutes. If you are working with vapor at its dew point or wet gas, there is a problem. If the flow transmitter is located below the orifice flanges, you will have to wait until the impulse lines refill with liquid. Open valve B, and close valves A and C. Now wait until the flowmeter indication stops changing. It ought to go back to zero, if the lines are refilled.

10.4.5 Zeroing Out a Flowmeter

The indicated flow of acetic acid is 9000 liters per day. The instrument technician checks the flowmeter to see if it has drifted by opening valve B with A and C closed (see Fig. 10.7). It should go back to zero—but a reading of 2000 liters per day is noted. The full range on the flowmeter is 10,000 liters per day. What is the real flow rate of the acetic acid? The answer is not 7000 liters. Why? Because flow varies with the square root of the orifice plate pressure drop. To calculate the correct acetic acid flow:

- $9000^2 - 2000^2 = 77,000,000$
- $(77,000,000)^{1/2} = 8780$ liters per day

The lesson is that near the top end of its range, the indicated flow is likely to be accurate, even if the meter is not well zeroed or the measured ΔP is not too accurate. On the other hand, flowmeters using orifice plates cannot be very accurate at the low end of their range, regardless of how carefully we have zeroed them. Digitally displayed flows also follow this rule.

10.5 Temperature

10.5.1 Temperature Indication

Figure 10.9 shows an ordinary thermowell-and-thermocouple assembly. The thermocouple junction consists of two wires of different metals. When this junction of the wires is heated, a small electric current, proportional to the junction temperature, is produced. Different metal wires make up the three most common junctions: *J*, *H*, and *K*.

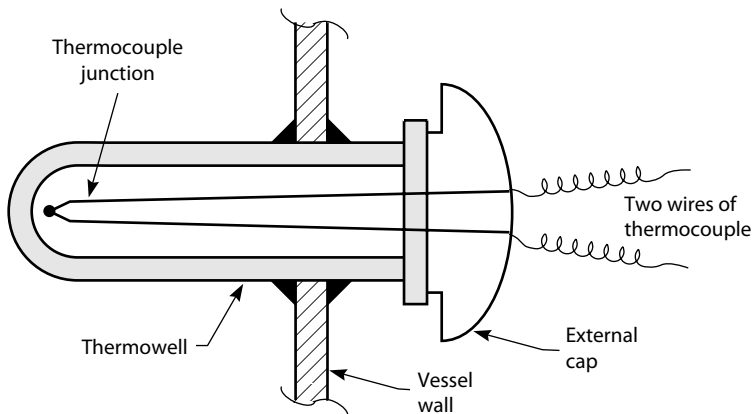


FIGURE 10.9 Thermocouple assembly.

It is not uncommon for a thermocouple, regardless of the type of junction, to generate too low a temperature signal.

If the exterior of the thermowell becomes fouled, the indicated temperature, generated by the thermocouple, will drop. The problem is that the external cap of the thermowell assembly radiates a small amount of heat to the atmosphere. Normally, this has a negligible effect on the indicated temperature. However, when the process temperature is 600 to 800°F, the thermowell is in a vapor phase, and the thermowell becomes coated with coke. I have seen the indicated temperature drop by 40°F below its true value. To verify that fouling of a thermowell is a problem, place a piece of loose insulation over the exterior thermowell assembly. If the indicated temperature rises by 5 or 10°F, then fouling on the outside of the thermowell is proved.

10.5.2 Short Thermowells

For a thermocouple to read correctly, it should be fully inserted in a thermowell and the thermowell itself should extend several inches into the process liquid. If the process stream is a vapor, which has poorer heat-transfer properties than do liquids, the thermowell, especially if the external insulation is poor, should extend more than 6 inch into the process flow. To check the length of the thermowell, unscrew the thermocouple assembly and pull it out, then simply measure the length of the thermocouple. This is also a good opportunity to verify the control-room reading with a portable temperature probe or a glass thermometer inserted in the thermowell. About 5 percent of the T1 points used are not located where the unit piping and instrumentation diagrams (P&IDs) show them to be. Then, pulling the thermocouple from a point in the process sometimes causes an unexpected drop to ambient temperature at an entirely unexpected T1 location.

10.5.3 Safety Note

Figure 10.9 is somewhat misleading. It implies that most thermowells are welded in place. Typically, thermowells are screwed into a 1-inch nozzle in the vessel wall.

I talked in the preceding section about unscrewing a thermocouple. This is perfectly safe. The same comment applies to extracting a screwed-in dial thermometer from a thermowell. However, sometimes less experienced people have made a deadly error in unscrewing a thermowell instead of a thermocouple.

Unscrewing a thermowell opens up the process fluid directly to the atmosphere. I myself once almost caused the death of a young engineer in Chalmette, Louisiana, when I told him to unscrew a thermowell. But I meant to say a thermocouple on a 300-psig, 500°F diesel oil pipe. He had the thermowell halfway out when I stopped him.

"Next time," I shouted, "pay attention to what I'm thinking, not what I'm saying."

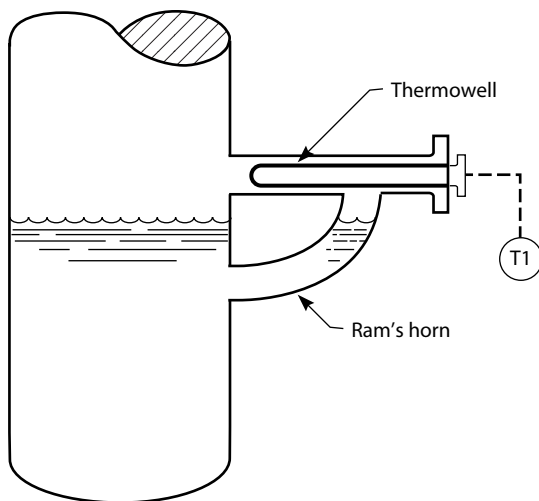


FIGURE 10.10 Using temperature to sense a liquid level.

10.5.4 Ram's-Horn Level Indication

A too short thermowell in the side of a vessel will increase in temperature when the liquid level rises to submerge the thermowell. Figure 10.10 illustrates a common method of exploiting this phenomenon. This is the ram's-horn interface-level indicator. The thermowell extends to the vessel wall only and is poorly insulated. The curved pipe below the thermowell permits liquid to drain out of and through the pipe enclosing the thermowell. Usually, three or four such ram's horns are vertically set 18 inch apart. A sudden temperature increase at a ram's horn is a foolproof method of detecting a rise in a liquid level. Especially in fouling or plugging service, I have seen this simple, archaic method of level indication succeed when all else fails.

If two temperature readings from the same point in the process disagree, the chances are that a temperature indication is more accurate than a temperature-control signal. The temperature signal used for control has usually been converted from its direct milliamp output to operate a control station. The temperature indication is generated right from the junction of the thermocouple, and hence there is less chance for error.

Reference

1. N. P. Lieberman, *Troubleshooting Process Plant Control* (Wiley, 2009).

CHAPTER 11

Packed Towers: Better Than Trays?

Packed-Bed Vapor and Liquid Distribution

The very first continuous distillation column was the Patent Still used to produce Scotch whiskey in the 1830s. It had 12 bubble-cap trays with weirs, downcomers, tray decks, and bubble caps with internal risers. Current trayed towers are quite similar. As most distillation towers have always been trayed rather than packed, one would have to conclude that trayed towers must have some sort of inherent advantage over packed towers. And this is indeed true in a practical sense, even though, in theory, a packed tower has greater capacity and superior separation efficiency than a trayed column.

11.1 How Packed Towers Work

The original packed towers used *Raschig rings*, hollow ceramic cylinders, typically 1 inch outer diameter (OD), 0.75 inch inner diameter (ID), and 1 inch long. A state-of-the-art packing is called *structured packing*. This material is made of thin sheets of crimped or corrugated metal, with small holes, fastened together with thin bars. The metal selected for the sheets is typically 316 stainless steel.

The two features that should be maximized in packed beds are:

- *Open area*—the average percentage of the cross-sectional area of the tower not blocked by the packing, and hence available for the flow of vapor and liquid.
- *Wetted surface area*—the number of square feet of packing surface area available for vapor-liquid contacting, per cubic foot of tower volume.

The greater the open area of packing, the greater the capacity of a tower. The greater the wetted surface area of a packing, the greater the separation efficiency of the tower. For example, a packing consisting of empty space would have lots of capacity but awful separation efficiency. A packing consisting of a fine sand would have great separation efficiency but very low capacity. So the selection of packing for a column is a compromise between maximizing open area and maximizing the wetted surface area.

Structured packing has about 50 percent more open area than Raschig rings and two or three times their wetted surface area. Hence, structured packing has largely replaced packing in the form of rings in many packed towers.

In any type of packed tower, the liquid or internal reflux drips through the packing and forms a thin film of liquid on the surface of the packing. Vapor percolates up through the packing and exchanges heat and molecules with the thin film of liquid on the surface of the packing.

In a trayed tower, vapor-liquid contact occurs only 5 or 6 inches above the tray deck, and the majority of the tower's volume is not used to exchange heat or mass between vapor and liquid. In a packed tower, the entire packed volume is used for this vapor-liquid contacting.

In a trayed tower, the area used for the downcomer that feeds the liquid to a tray and the area used for draining liquid from a tray are unavailable for vapor flow. In a packed tower, the entire cross-sectional area of the tower is available for vapor flow.

This certainly makes it seem as if packing is vastly superior to trays, and even if this was not true in the past, with the advent of structured packings in the 1980s it ought to be true now. But as a process design engineer, whenever possible I still specify trays rather than packing. Why?

11.1.1 Liquid Distribution

Each tray in a tower is inherently a vapor-liquid *redistributor*. The holes in the tray decks redistribute the vapor flowing up the tower. The outlet weir, or more exactly the crest height of the liquid overflowing the weir, forces the liquid to flow evenly across the tray. Even if the weir height is uneven on a tray, and liquid flow is distorted on that particular tray, the liquid will be properly redistributed on the tray below.

Unfortunately, packing does not redistribute liquid or internal reflux. Unless the initial reflux distribution is good, the liquid flow distribution through the entire packed bed will be poor. Figure 11.1 shows a common orifice plate liquid distributor. Vapor flows up through the large chimneys, and liquid drains through the smaller distribution holes in the tray deck.

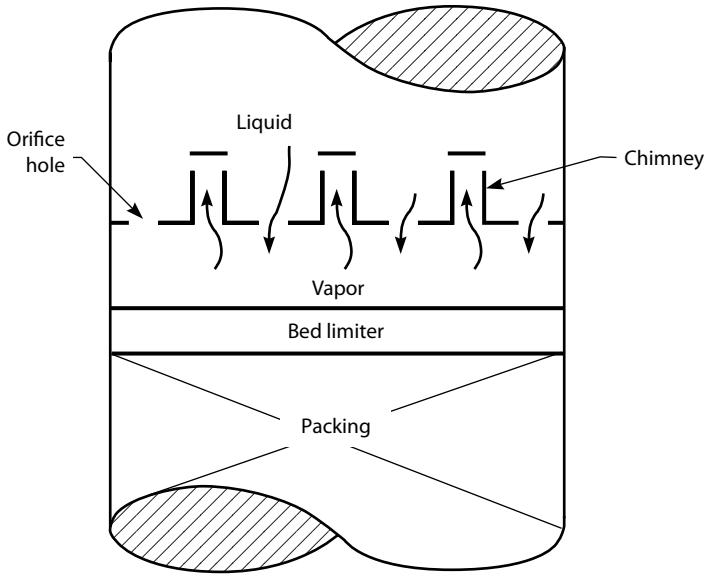


FIGURE 11.1 Orifice plate liquid distributor.

To calculate the height of liquid on this chimney tray, we have to add two factors:

- The pressure drop of the vapor flowing through the chimneys. The liquid on the tray has to develop enough liquid head to flow against the higher pressure below the tray.
- The orifice hole pressure drop. This is the liquid head, which has to be converted to velocity as the liquid flows through the orifice holes on the tray deck.

The pressure drop of the vapor flowing through the chimneys is

$$\Delta P_v = K_c \frac{D_v}{D_l} (V_g)^2$$

where ΔP_v = pressure drop of the vapor flowing through the chimneys and hats, in inches of clear liquid

K_c = an orifice coefficient for a chimney plus a chimney hat, typically 0.6 to 0.9

D_v = density of vapor

D_l = density of liquid

V_g = velocity of vapor passing up through the chimneys, ft/s

The orifice hole pressure drop is

$$\Delta P_L = K_0 (V_L)^2$$

where ΔP_L = pressure drop of the liquid flowing through the orifice holes, in inches of clear liquid

K_0 = an orifice coefficient for a hole in metal plate, typically 0.4 to 0.6

V_L = velocity of liquid draining through the orifice holes, ft/s

The sum of $\Delta P_V + \Delta P_L$ is the total height of liquid on the chimney tray. Let us assume that this total height is 6 inches of liquid:

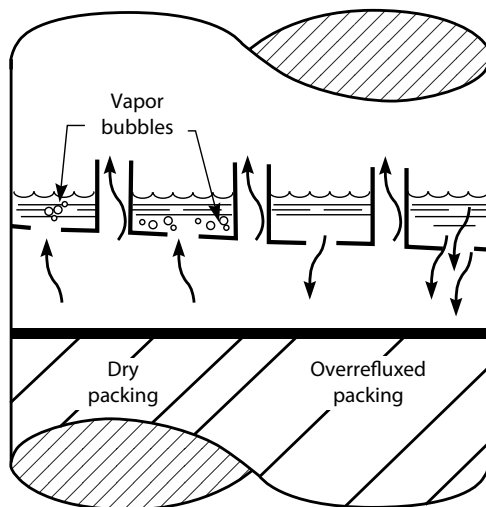
- $\Delta P_V = 5.1$ inch
- $\Delta P_L = 0.9$ inch

Let's further assume that the orifice plate distributor is 1 inch *out of level*. This could easily happen in a 14-ft 0-inch-ID tower. Figure 11.2 shows the results. The flow of internal reflux or liquid through the higher portion of the tray deck falls to zero. Worse yet, vapor starts to bubble up through any orifice hole that is not submerged by at least 5.1 inch of *clear liquid*. But, of course, once vapor bubbling begins, we do not have clear liquid in this region of the tray deck.

Meanwhile, all the liquid flow must drain through the lower portion of the tray. The net result, as can be seen in Fig. 11.2, is dry packing in one portion of the tower and overrefluxing in the adjacent portion. This is called *vapor-liquid channeling*, and this is the root cause of poor fractionation efficiency in any tower.

One way out of this problem is to increase ΔP_L the pressure drop of the liquid flowing through the orifice holes. This could be done by increasing the orifice hole liquid velocity. We could drill fewer orifice holes. Unfortunately, this would decrease the number of *drip points* per square foot of tower area (6 to 10 is a good target). This would

FIGURE 11.2
Orifice plate distributor not level.



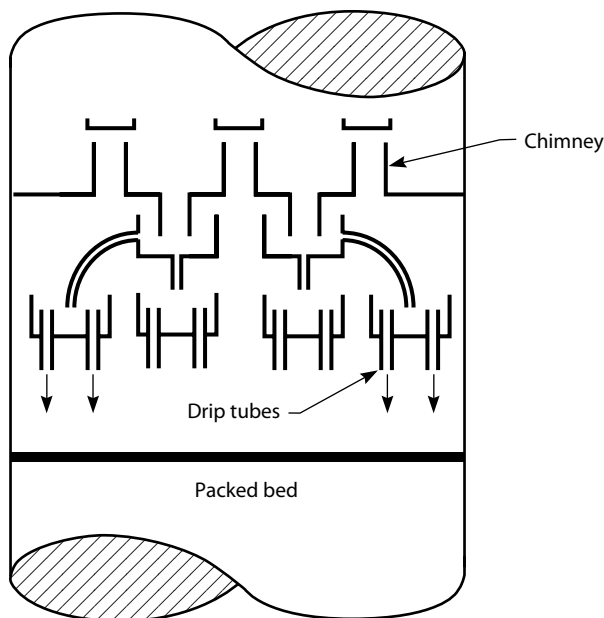


FIGURE 11.3 Modern liquid distributor.

reduce vapor-liquid contacting efficiency. Or we could have smaller orifice holes. But too small a hole would probably plug with corrosion products.

There are ways around these problems. Figure 11.3 is a simplified sketch of a multipan liquid distributor using raised, slotted, guided drip tubes employing an initial distribution header. It is beyond the scope of this text to describe how this works. But I will say that such liquid distributors are:

- Expensive to build
- Tricky to design—especially for small liquid flows
- Difficult to install correctly
- Occupy quite a few feet of tower vertical height

11.1.1.1 Trade-Off between Efficiency and Capacity

In a trayed tower, we can increase tower capacity by reducing the number of trays and increasing the *tray spacing*. But reducing the number of trays in turn reduces separation efficiency. In a packed tower, we can increase tower capacity by using larger packing and increasing the open area (or a bigger crimp size for structured packing). But the reduction in the wetted area of the packing will reduce separation efficiency. Installing a properly designed, efficient liquid distributor, such as the

one shown in Fig. 11.3, will definitely enhance separation efficiency, but at the expense of the vertical height of the tower, which could otherwise be devoted to a greater height of packing.

11.1.2 Vapor Distribution

Figure 11.4 shows a modern, narrow-trough, liquid collector–vapor distributor chimney tray. While the initial vapor distribution through a packed bed is not quite so critical or difficult as the liquid distribution, it is still important.

A properly designed valve or sieve tray will act as a vapor redistributor. Thus, poor initial vapor distribution will only lessen the efficiency of the bottom tray. But if a packed-bed vapor distributor does not work properly, vapor channeling will be promoted through the entire bed.

The narrow-trough vapor distributor shown in Fig. 11.4 is intended to disperse the vapor evenly across the bottom of the packed bed. The width of the chimney does not exceed 6 in. The older-style

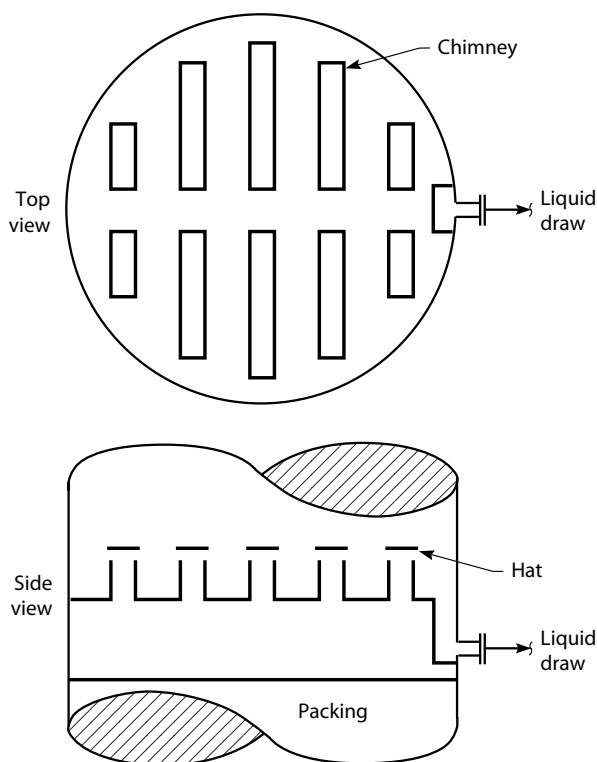


FIGURE 11.4 Modern vapor distributor.

chimney trays, which may have had a few large round or square chimneys, reduced the separation efficiency of the packing. To work properly, the vapor distributor has to have a reasonable pressure drop in comparison to the pressure drop of the packed bed. For example, if the expected pressure drop of a 12-ft packed bed is 10 inches of liquid, the pressure drop of the vapor distributor ought to be about 3 to 4 inches of liquid.

In practice, the most common reason why vapor flow is maldistributed as it escapes from a well-designed chimney tray is distortion of the hats. I believe this is due mostly to workers stepping on the hats during unit turnarounds. The hat support brackets are designed to support a 20-lb hat, not a 200-lb welder.

11.2 Maintaining Functional and Structural Efficiency in Packed Towers

11.2.1 Pressure Drop

A packed tower can successfully fractionate with a very small pressure drop, as compared to a tray. For a modern trayed tower, to produce one single theoretical tray worth of separation (that's like a single, 100-percent efficient tray), a pressure drop of about 6 inch of liquid is needed. A bed of structured packing can do the same job with 1 inch of liquid pressure drop, even when allowing for the vapor distributor. In low-pressure fractionators, especially vacuum towers used to make lubricating oils and waxes, this can be of critical importance.

Packed beds also seem to have a better turndown capability than valve or sieve trays at low vapor flows. On the other hand, many packed fractionators seem quite intolerant of reduced liquid or reflux flow rates. This is typically a sign of an improperly designed distributor in the packed fractionator.

The problem we have just discussed—poor fractionation efficiency due to inadequate vapor and liquid initial distribution—is rather similar to tray deck dumping in trayed fractionators. And, just like trays, packed towers are also subject to flooding.

11.2.2 Flooding in Packed Towers

To understand what is meant by flooding in a bed of packing, one ought to first understand what is meant by the idea of *holdup*.

Let's imagine we are operating the air-water scrubber in the Unit-Ops Lab at Cooper Union School of Engineering (Fig. 11.5). The water is circulating from the bottom to the top of the tower. The water level in the bottom is 2 ft. The height of the packed bed is 20 ft. At 3:30 p.m., Professor Liebskind shuts off our air compressor and the water circulation pump. As the water, which had been held up on the packing by

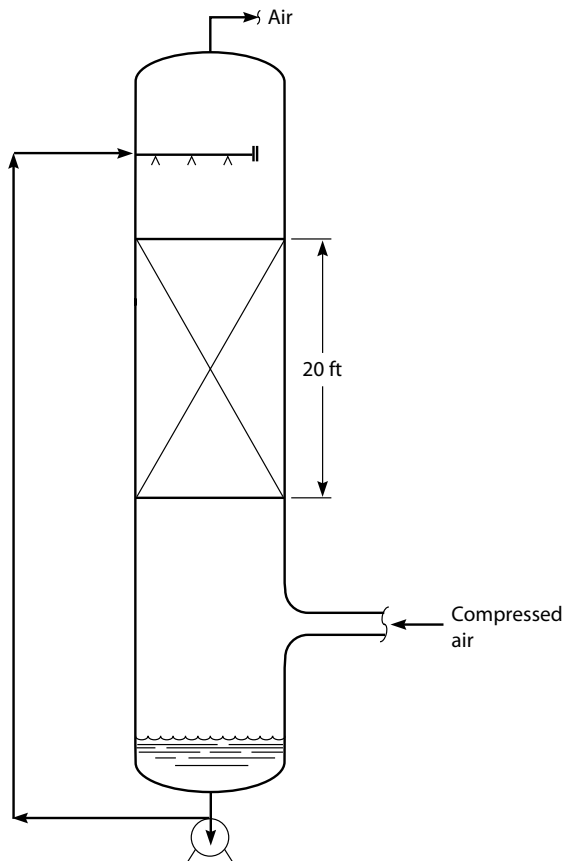


FIGURE 11.5 Measuring holdup in a packed bed.

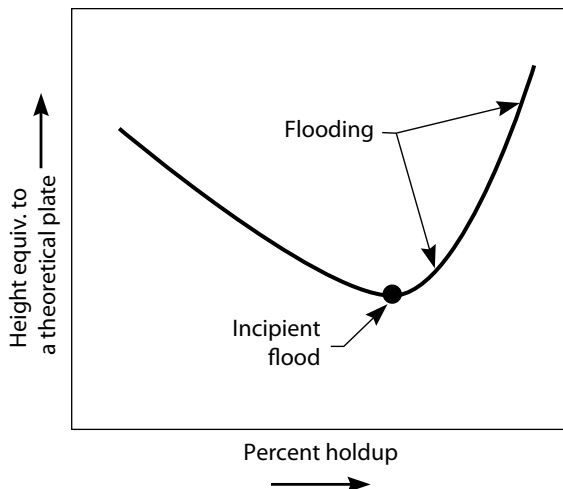
the flowing air, drains down, the water level in the bottom of the tower rises from 2 to 5 ft. We say, then, that the holdup of the packing was 15 percent $[(5 \text{ ft} - 2 \text{ ft}) \div 20 \text{ ft} = 15 \text{ percent}]$.

If the liquid holdup is too low, fractionation efficiency will be bad. We say that the *height equivalent to a theoretical plate* (HETP) will be high. If the liquid holdup is too high, fractionation efficiency will also be poor. We again say that the HETP will be high. This idea is expressed in Fig. 11.6. When the holdup rises above the point that corresponds to the minimum HETP, we can say that the packing is beginning to flood. The minimum HETP point in Fig. 11.6 corresponds to the point of an incipient flood, discussed in Chap. 3, for trayed towers.

For structured-type packing, a liquid holdup of 4 to 5 percent corresponds to this optimum packing fractionation efficiency. For 1-inch Raschig rings, this optimum holdup would be roughly 10 to 12 percent.

FIGURE 11.6

Holdup predicts
flooding.



There is a close relationship between percent liquid holdup and pressure drop, as expressed in inches of liquid per foot of packing. If the pressure drop per foot of packing is 1.2 inch, then the liquid holdup is roughly 10 percent ($1.2 \text{ inch} \div 12 \text{ inch} = 10 \text{ percent}$). Unlike the perforated tray decks discussed in Chap. 3, there is no simple equation to predict the optimum pressure drop that would correspond to the minimum HETP for different types of packing. There is just too much diversity in both size and shape to generalize. I can, however, recall a delayed coking fractionator in Los Angeles with a 7-ft bed of structured packing. When the pressure drop per foot of packing reached 4 inch (due to coke formation), drainage of liquid through the bed ceased.

11.2.2.1 Packed-Bed Supports

A simple support for a bed of 1.5-inch pall rings might consist of a bar grid with 0.25-inch-thick bars set on 1-inch center. The grid itself (see Fig. 11.7) would be supported by two 6-inch-wide I-beams. The open area of each component is

- 1.5-inch pall rings: 80 percent
- Grid support: 75 percent
- I-beam support: 90 percent

Considering that the rings, the grid support, and the I-beam are all in close physical contact, what is the open area at their mutual interface? (Answer: $80 \text{ percent} \times 75 \text{ percent} \times 90 \text{ percent} = 54 \text{ percent}$.) This means that there will be a restriction to vapor and liquid flow at the grid support, which will promote flooding at the bottom of the bed. Naturally, this would cause the entire packed bed to flood.

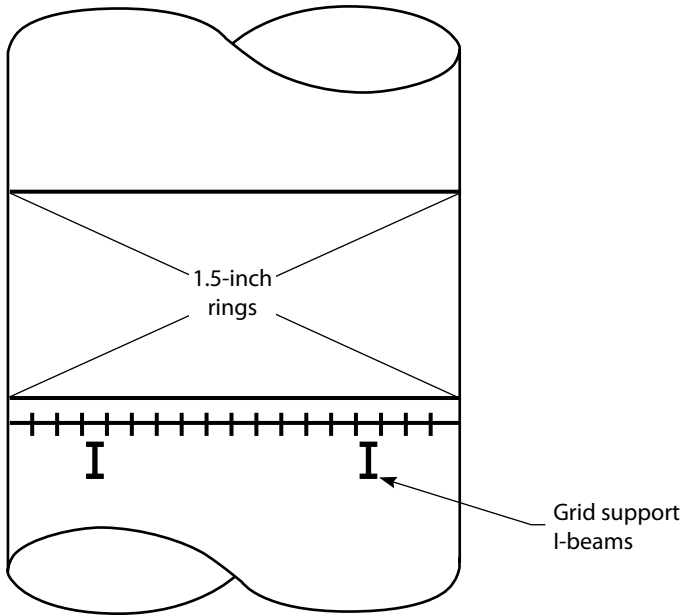
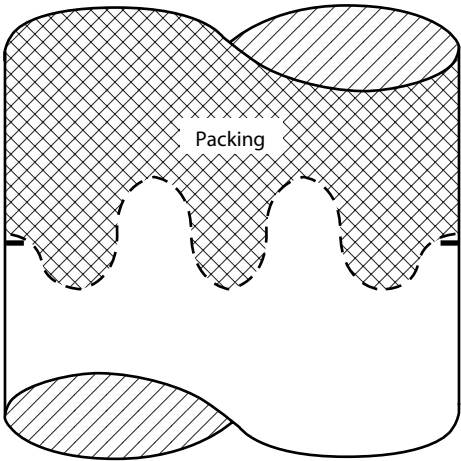


FIGURE 11.7 A restrictive packing support will cause flooding.

To avoid such flooding, the designer uses the corrugated bed support shown in Fig. 11.8. This grid will have over 100-percent open area based on the tower cross-sectional area. Crushed packing lying on even this excellent type of bed support can, however, cause flooding if the crushed packing has lost most of its open area.

FIGURE 11.8
Superior bed support.



11.2.3 Packing Holddowns

During routine operations, the weight of a bed of packing is sufficient to keep it in place. But if there is a pressure surge during start-up, the packed bed can be forcefully pushed against the underside of the liquid distributor. If this liquid distributor is a spray header, as illustrated in Fig. 11.9, the spray bar arms can be bent. To prevent this, a packing top holddown grid is required. This holddown grid must be firmly attached to the vessel wall. For example, in a 12-ft-ID tower, the holddown grid is designed to resist an uplift force due to a pressure surge of 0.5 psi. What is the total uplift force?

$$\text{Answer: } 0.5 \text{ psi} \times 144 \frac{\text{inch}^2}{\text{ft}^2} \times \frac{3.14}{4} \times 12^2 = 8000 \text{ lb}$$

Figure 11.10 illustrates a problem introduced when a packing holddown was installed at the wrong elevation. This tower was designed to purify acetic acid by distilling overhead the lighter formic acid. But at even 50 percent of the design reboiler duty, the tower flooded, and acetic acid was carried over into the formic acid. The problem was that the perforated plate holddown used to keep the 1.5-inch Teflon rings from floating up through the feed chimney tray distributor was located immediately below these chimneys. What, then, was the open area at the interface of the chimney feed distributor, the 1.5-inch rings, and the holddown plate?

- Open area of chimney feed distributor = 40 percent
- Open area of rings = 70 percent

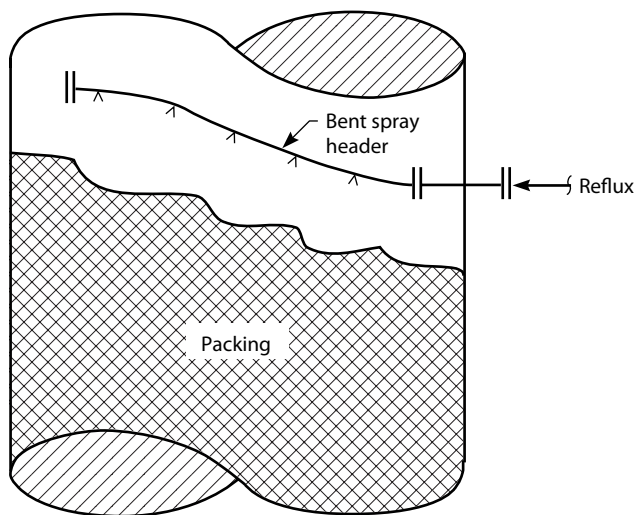


FIGURE 11.9 Why packing holddown grids are important.

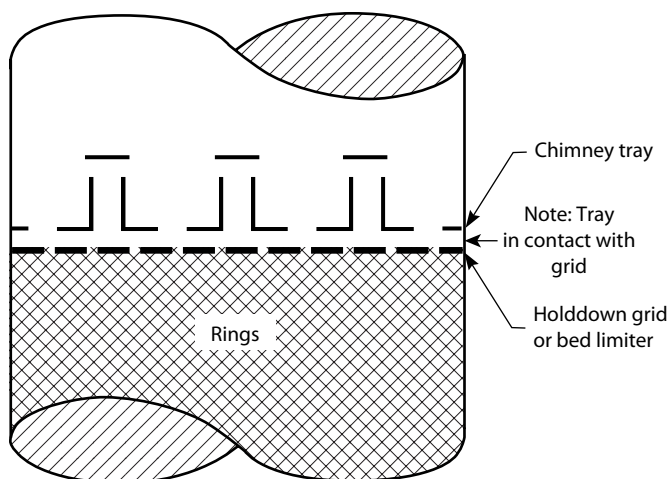


FIGURE 11.10 Flooding caused by mislocation of holddown grid.

- Open area of holddown plate = 30 percent
(Answer: 40 percent \times 70 percent \times 30 percent = 8.4 percent.)

As a result, the tower flooded above the holddown plate. To fix this problem, the packing holddown was dropped 15 inch below the chimney tray. As a result, the tower fractionated properly.

Incidentally, this incident was a multimillion-dollar mistake. It was not, however, a design error. The installation crew had simply found a “cheaper” location to install the holddown grid by using the same support as the chimney feed distributor.

11.2.4 Crushed Packing

One of the great disadvantages of packing is the inability to properly inspect the installation. I well remember one coal gasification project in Chicago, where the off-gas scrubber flooded because of a plastic bag left inside a packed bed of pall rings. On another tower, flooding of a bed of structured packing occurred at 50 percent of design. Only when the packing was removed for inspection was the problem revealed. Workers had previously stepped on and crushed intermediate layers of packing. Then, probably without realizing that they had reduced the open area of the structured packing by 50 percent, they set additional layers of structured packing on top of the damaged layers.

The situation on a trayed tower is far more controllable. Even after the tray assembly is complete, the process engineer, using only a crescent wrench, can open a few tray deck manways and double-check the tower internals. But to pull 6 ft of packing out of a tower is a major undertaking. Also, trays can be stepped on without damage. Beam supports and 40-lb air guns can be dragged across a sieve tray

with impunity. Packings—rings or structured—require special handling during installation or turnarounds. To be frank, all packing installations that I have seen have been damaged to some extent.

To summarize, the chief disadvantages of packed towers, as compared to trays, are

- Difficulty of achieving proper initial liquid distribution, especially at small liquid flow rates
- The need for a proper vapor distributor to achieve proper initial vapor distribution
- Possible restrictions to vapor flow from the packing grid support or holddown
- Problems in inspecting the final installation

11.3 Advantages of Packing vs. Trays

Often, we remove heat from a tower at an intermediate point by use of a *pumparound* or circulating reflux. Figure 11.11 is a sketch of such a pumparound. In many towers, the liquid flows in the pumparound

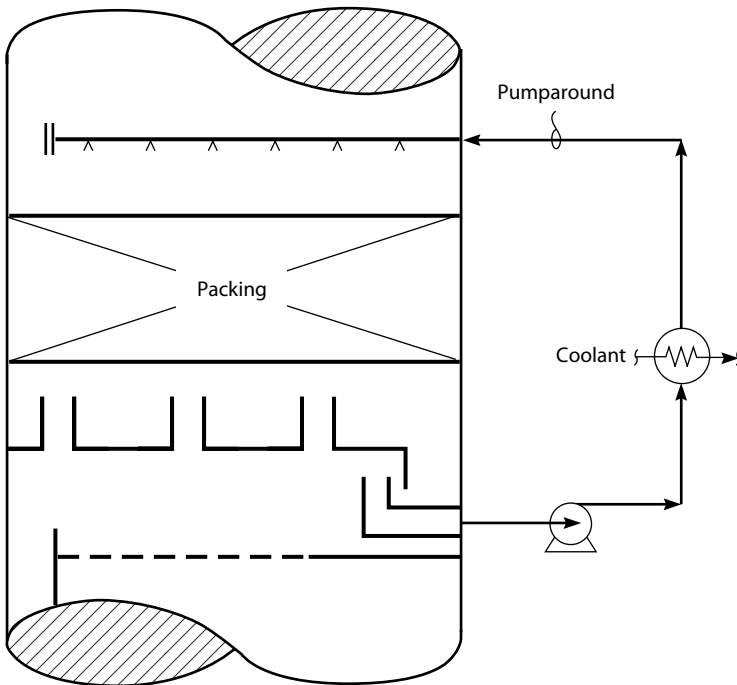


FIGURE 11.11 Packing is often used in pumparound sections of trayed towers.

section are greater than in the other sections, which are used for fractionation. That is why we are often short of capacity and initiate flooding in the pumparound or heat-removal section of a column.

Since we do not rely on pumparounds to fractionate—but just to remove heat—good vapor-liquid distribution is not critical. A bed of 4 or 5 ft of structured packing is often, then, an excellent selection for the pumparound section of a tower. The capacity of such a bed potentially has a 30 to 40 percent advantage over trays.

Once a tower's diameter is 3 ft or less, trays become difficult to install, due to the confined working area. Also, the tray support ring appreciably cuts down on the tower's cross-sectional area. Dumped-type packings (rings, saddles, broken beer bottles) do not present such problems. Also, for high liquid flows in a smaller-diameter tower (less than 5 ft), a well-designed packed tower will have a very substantial capacity advantage over a trayed tower. In general, the lower the pressure of a tower, the greater the advantage of using packing over trays.

Finally, the pressure drop of the vapor flowing through a packed tower will be an order of magnitude less than through a trayed tower. For vacuum distillation service, this is often of critical importance.

Reference

1. N. P. Lieberman, *Process Design for Reliable Operations*, 1st edition, Gulf Publishing, 1983. Current 3rd Edition, Lieberman Books, 2008.

CHAPTER 12

Steam and Condensate Systems

Water Hammer and Condensate Backup Steam-Side Reboiler Control

In this chapter we discuss process equipment such as

- The steam side of reboilers
- Steam turbine surface condensers
- Condensate recovery systems
- Deaerators
- The steam side of shell-and-tube steam preheaters

A few of the nasty features of this sort of process equipment we discuss are

- Condensate backup
- Accumulation of carbon dioxide
- Steam hammer
- Blown condensate seals

12.1 Steam Reboilers

When considering the steam side of steam-heated reboilers, it is best to think about the reboiler as a steam condenser. The steam, at least for a conventional horizontal reboiler, is usually on the tube side of the exchanger, as shown in Fig. 12.1. The steam is on the tube side,

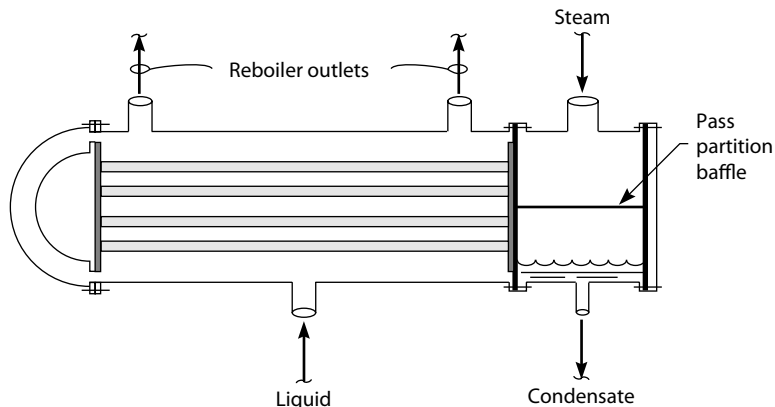


FIGURE 12.1 A shell-and-tube steam reboiler.

because the shell side was selected for the process fluid. If the reboiler is a thermosyphon, or natural-circulation reboiler, then low-process-side pressure drop is important. For a horizontal reboiler, it is easiest to obtain a low pressure drop for the fluid being vaporized by placing it on the shell side.

The steam enters through the top of the channel head of the reboiler. Any superheat in the steam is quickly lost to the tubes. Superheating steam does very little in increasing heat-transfer rates in a reboiler. Actually, when considering the temperature difference between the steam and the process fluid, it is best to use the *saturated steam* temperature as the real temperature at which all the heat in the steam is available. For example, assume the following steam flow to a reboiler:

- 1000 lb/h of 100-psig steam
- 400°F steam inlet temperature
- 300°F condensate outlet temperature

The 100-psig steam condenses at approximately 320°F. The heat available from the steam is

- Desuperheating = $(400^{\circ}\text{F} - 320^{\circ}\text{F}) \times [0.5 \text{ Btu}/(\text{lb})(^{\circ}\text{F})] \times 1000 \text{ lb/h} = 40,000 \text{ Btu/h}$
- Condensing = $900 \text{ Btu/lb} \times 1000 \text{ lb/h} = 900,000 \text{ Btu/h}$
- Subcooling = $(320^{\circ}\text{F} - 300^{\circ}\text{F}) \times [1.0 \text{ Btu}/(\text{lb})(^{\circ}\text{F})] \times 1000 \text{ lb/h} = 20,000 \text{ Btu/h}$
- Total reboiler duty = 960,000 Btu/h

This calculation is typical in that 94 percent of the heat is liberated at the 320°F condensing temperature of the saturated steam. Another way of stating the same idea is that a steam reboiler depends on *latent-heat transfer* and not on *sensible-heat transfer*.

12.2 Condensing Heat-Transfer Rates

When a vapor condenses to a liquid, we say that the latent heat of condensation of the vapor is liberated. In a steam reboiler, this liberated heat is used to reboil the distillation tower. When a vapor, or more commonly a liquid, cools, we say that its sensible heat is reduced. For a small or slight temperature change, the change in latent heat might be large, while the change in sensible heat will be very small.

Heat exchange provided by sensible-heat transfer is improved when velocities are higher. Especially when the heating fluid is on the tube side of an exchanger, sensible-heat-transfer rates are always increased by high velocity.

Heat exchange provided by latent-heat transfer is improved when velocities are lower. It is my experience that this loss of heat transfer at high velocity is quite large when steam is flowing through the tube side of an exchanger. Theoretically, this happens because the condensing film of steam is blown off the tube surface by the high vapor velocity.

This improved heat-transfer rate, promoted by low velocity, applies not only for condensing steam but also for condensing other pure-component vapors. And since condensation rates are favored by low velocity, this permits the engineer to design the steam side of reboilers and condensers in general for low-pressure drops. For example, if we measured the pressure above the *channel head pass partition* baffle shown in Fig. 12.1, we would observe a pressure of 100 psig. The pressure below the channel head pass partition baffle would typically be 99 psig.

12.2.1 Blown Condensate Seal

Figure 12.2 shows a common type of reboiler failure. The steam trap on the condensate drain line has stuck open. A *steam trap* is a device intended to open when its float is lifted by water. The steam trap remains open until all the water drains out of the trap. Then, when there is no more water to keep the trap open, it shuts. But if the float sticks open, steam can blow through the steam trap. This is called a *blown condensate seal*. The average vapor velocity through the tubes will then accelerate. If the steam trap is passing a lot of steam, the velocity of steam in the tubes increases a bit. I have seen this often. Blowing the condensate seal due to a faulty steam trap causes a loss in reboiler duty of 50 percent or more. This is due to the increased steam tube-side velocity, reducing the rate of condensation of the steam, and hence reducing the rate of liberation of the latent heat of condensation of the steam.

12.2.2 Condensate Backup

What would happen to a steam reboiler if the float in the steam trap became stuck in a partly closed position, or if the steam trap were too small? Water—that is, steam condensate—would start to back up into

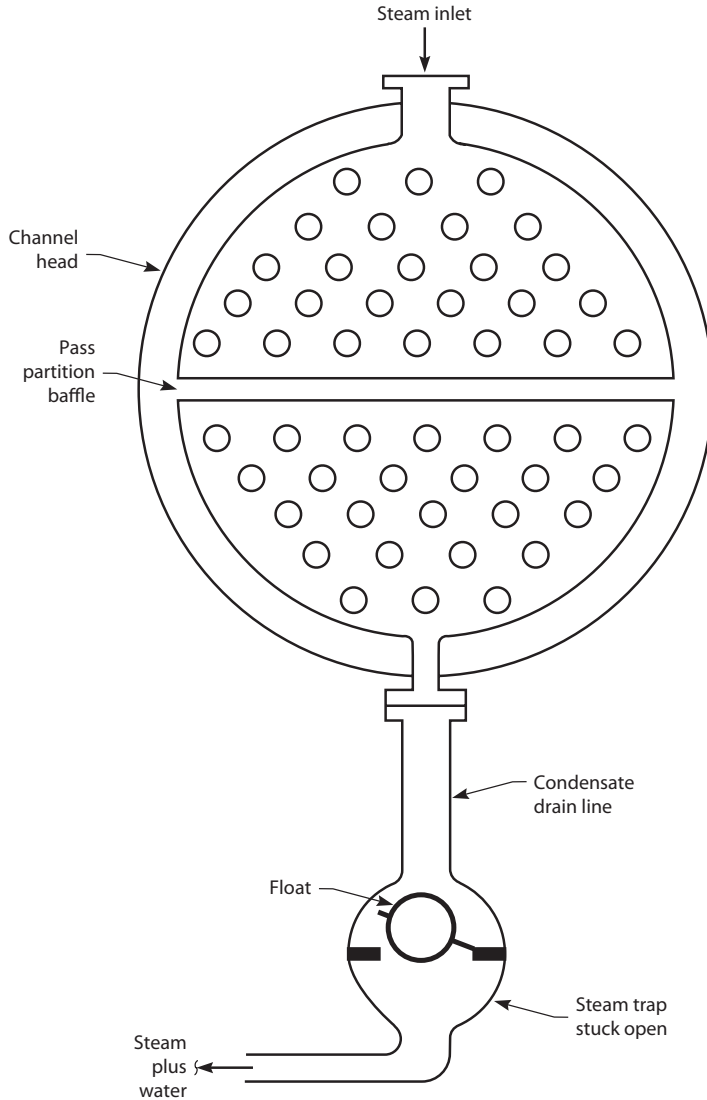


FIGURE 12.2 Blown condensate seal.

the channel head of the reboiler, as shown in Fig. 12.3. The bottom tubes of the reboiler bundle would become covered with water. The number of tubes exposed to the condensing steam would decrease. This would reduce the rate of steam condensation and also the reboiler heat duty.

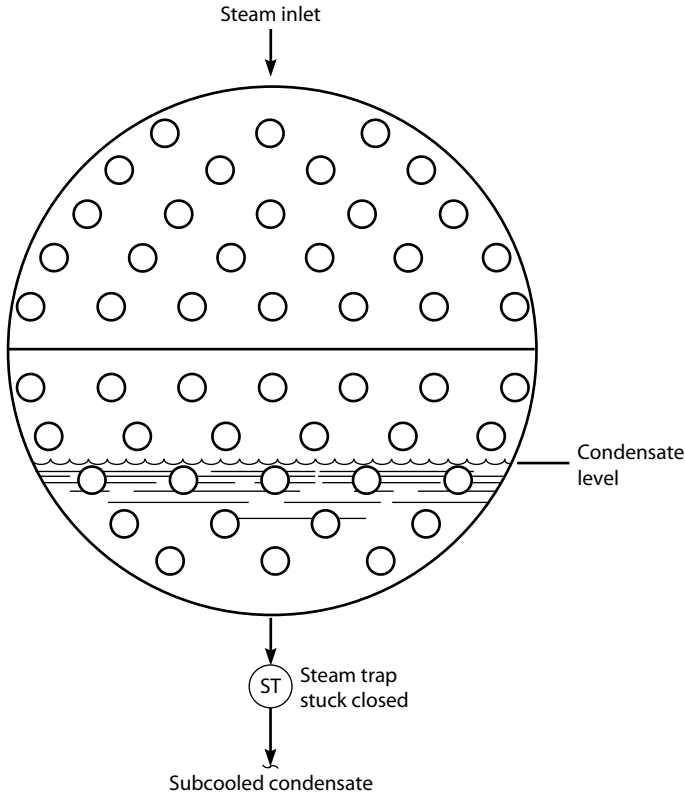


FIGURE 12.3 Effect of condensate backup.

Meanwhile, the tubes covered with stagnant water would begin to cool. The steam condensate submerging these tubes would cool. This cooled water would be colder than the saturation temperature of the condensing steam. The tubes would then be said to be submerged in *subcooled* water.

We can see, then, that either condensate backup or blowing the condensate seal will cause a steam reboiler to lose capacity. If you think either of these two problems could cause a loss in reboiler duty, try opening the bypass around the steam trap. If the reboiler duty goes up, the problem was condensate backup. If the reboiler duty goes down, the problem might be a blown condensate seal. If it looks like a blown condensate seal problem, close the steam trap bypass. Then partially close the valve downstream of the steam trap. If this increases the reboiler duty, a blown condensate seal failure is proved.

12.3 Maintaining System Efficiency

12.3.1 Steam Flow Control

The flow of steam to a reboiler can be controlled by using a control valve on either (1) the *steam inlet line* or (2) the *condensate outlet line*.

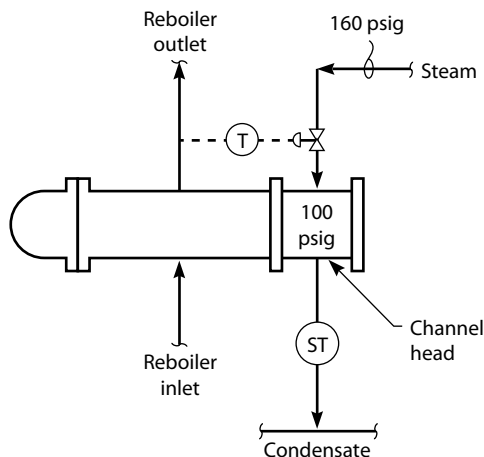
Figure 12.4 shows a control valve on the steam inlet line. The rate of steam flow to the reboiler is not really controlled directly, however, by this control valve. The actual rate of steam flow to the reboiler is controlled by the rate of condensation of the steam inside the tubes. The faster the steam condenses, the faster it flows into the channel head. The function of the control valve is to reduce the steam pressure in the channel head of the reboiler. For example, in case 1, described below:

- 160-psig steam supply header pressure.
- ΔP across control valve = 60 psi and the control valve is 40 percent open.
- 100-psig steam pressure, condensing at 320°F, in the channel head.
- Shell-side reboiler temperature of 240°F.
- Steam flow = 10,000 lb/h.
- Temperature difference between the condensing steam and the boiling process liquid is then (320°F – 240°F) = 80°F. This is called the *temperature-difference driving force*, or ΔT .

Now the rate or heat duty of steam condensation is termed Q :

$$Q = A \times U(\Delta T)$$

FIGURE 12.4
Varying channel
head pressure
controls heat
input.



where A = surface area of the tubes, which are exposed to the condensing steam, ft^2

U = heat-transfer coefficient—a constant, describing the rate of condensation of steam, on those tubes exposed to the condensing steam ($\text{Btu}/[(\text{h})/(\text{ft}^2)(^\circ\text{F})]$)

ΔT = temperature difference between the shell and tube sides, $^\circ\text{F}$

Q = heat-exchanger duty, Btu/h (for the preceding equation in case 1; 10,000,000 Btu/h)

In case 2, the shell-side reboiler temperature rises from 240 to 280°F (one reason for such a rise in temperature could be an increase in tower pressure). Now $\Delta T = (320^\circ\text{F} - 280^\circ\text{F}) = 40^\circ\text{F}$. Looking at the equation on the previous page, it looks as if Q will drop in half to 5000 lb/h (which is about the same as 5,000,000 Btu/h). Thus, the flow of steam to the reboiler has been cut in half, even though the control valve position has not moved.

You might have noticed how we have used 1,000,000 Btu/h of heat interchangeably with 1000 lb/h of steam. This is approximately correct for low-pressure steam.

To consider a third case, we wish to maintain the original 240°F shell-side temperature, but to increase the steam flow from 10,000 to 15,000 lb/h . This will force the steam inlet control valve to open. As the control valve opens, the pressure in the channel head rises from 100 psig to the full steam header pressure of 160 psig. At this pressure, steam condenses at 360°F. The new ΔT is then $(360^\circ\text{F} - 240^\circ\text{F}) = 120^\circ\text{F}$. This new temperature driving force is 50 percent greater than the first case's driving force of 80 percent. Hence the rate of steam condensation also increases by 50 percent, from 10,000 to 15,000 lb/h .

In case 1, the steam inlet control valve was 40 percent open. In case 3, let's assume that the inlet control valve is 70 percent open. If we open the control valve to 100 percent, the steam flow will not increase at all. Why? Because once the steam pressure in the channel head rises to the steam header pressure, no further increase in steam flow is possible, regardless of the position of the inlet control valve.

12.3.2 Condensate Control

Once the steam pressure in the channel head of Fig. 12.4 falls to the pressure in the condensate collection header, the steam trap can no longer pass condensate. Water will back up in the channel head and water-log the lower tubes in the tube bundle. This will lead to unstable steam flow control. This is especially true if the steam supply pressure is less than 20 psig above the condensate collection header pressure.

It is better not to use a steam inlet control valve when using low-pressure steam. The channel head pressure will then always equal the steam header supply pressure. The flow of steam to the reboiler can

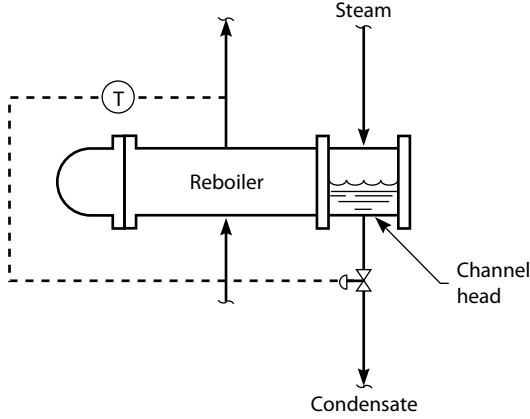


FIGURE 12.5 Condensate backup controls steam flow.

then be controlled only by raising or lowering the water level in the channel head, as shown in Fig. 12.5. This sort of control scheme will work perfectly well until the water level drops to the bottom of the channel head. If the condensate drain control valve then opens further in an attempt to increase steam flow into the reboiler, the condensate seal is blown, and the reboiler heat duty drops.

A better design is shown in Fig. 12.6. In this scheme, a condensate drum is used to monitor the level in the channel head. As the drum level is drawn down, the number of tubes in the reboiler exposed to

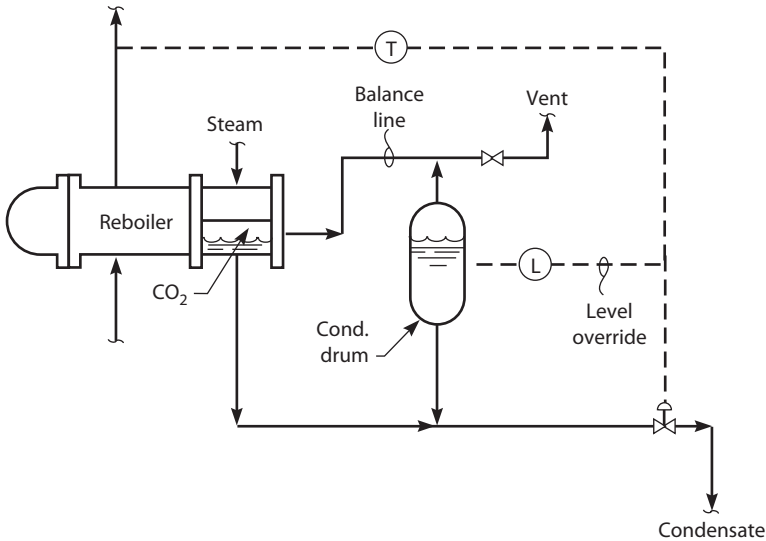


FIGURE 12.6 Venting below the pass partition baffle stops carbonic acid corrosion.

the condensing steam is increased. However, when the water level drops to the bottom of the channel head, the level sensor in the condensate drum assumes control over (i.e., overrides) the condensate flow controller and prevents loss of the condensate seal.

One important feature of Fig. 12.6 is the *condensate drum balance line*. Note that this line is connected below the channel head pass partition baffle. This ensures that the pressure in the channel head below the pass partition baffle and the pressure in the condensate drum are the same. If these two pressures are not identical, the level in the condensate drum cannot represent the level in the channel head. For this reason, never connect the condensate drum vapor space to either the steam supply line or the top vent of the reboiler's channel head.

12.4 Carbonic Acid Corrosion

Steam produced from demineralized water is largely free of carbonates. Steam produced from lime-softened water will be contaminated with carbonates that decompose in the boiler to carbon dioxide. As the steam condenses in a reboiler, the CO_2 accumulates as a non-condensable gas. This gas will be trapped mainly below the channel head pass partition baffle, shown in Fig. 12.6. As the concentration of CO_2 increases, the CO_2 will be forced to dissolve in the water:



That is, carbonic acid will be formed. Carbonic acid is quite corrosive to carbon steel. Reboiler tube leaks, associated with steam-side corrosion, are almost certainly due to carbonic acid attack.

Venting the channel head through the balance line shown in Fig. 12.6 will prevent an excessive accumulation of CO_2 . This is done by continuous venting from the top of the condensate drum. For every 10,000 lb/h of steam flow, vent off 50 lb/h of vapor through a restriction orifice placed in the condensate drum vent. This is usually cheaper than controlling reboiler steam-side corrosion with neutralizing chemicals.

Often the problem with CO_2 accumulation inside the channel head results not so much in corrosion, but in the loss of heat transfer. This is due to the tubes below the bottom pass partition baffle filling with non-condensable CO_2 gas. I have observed this problem even in plants using demineralized boiler feedwater. Venting from beneath the bottom pass partition baffle will restore heat transfer rates. Venting above the pass partition baffle or from the top of the channel head is futile. Unfortunately, most such channel head vents are installed on the top of the channel head, and thus are completely ineffective.

12.4.1 Channel Head Leaks

Varying the steam-to-condensate interface level to control the reboiler duty will promote steam leaks in the channel head-to-shell flanged

closure. This is caused by the thermal cycling and stresses that result from constantly varying the level of condensate in the channel head. However, when low-pressure steam (<60 psig) is used, this becomes a minor problem, which may be safely ignored.

When high-pressure steam (>100 psig) is used, rather significant leaks of hot condensate and steam can be caused by a variable condensate level in the channel head. For such higher-pressure steam sources, control of steam flow with condensate backup, as shown in Figs. 12.5 and 12.6, is best avoided.

12.5 Condensate Collection Systems

How much steam condensate is your plant recovering? Seventy percent is considered pretty good, and 30 percent is, by any standard, pretty awful. As condensate collection flows are rarely metered, here is a really good way to make such an overall measurement (ST = steam; TR = treated water):

1. Determine ST—the lb/h of steam raised in the whole plant.
2. Determine TR—the lb/h of softened or demineralized treated water flowing to the deaerators.
3. The percent of condensate recovery is then

$$100\% - 100\% (TR \div ST)$$

Loss of steam condensate to the plant's sewer is environmentally wrong and wastes money for water-treating chemicals and energy. The principal reasons why steam condensate is lost to the sewer are

- It creates steam or water hammer in the condensate collection system.
- Back-pressure from the condensate collection lines creates control difficulties in steam reboilers or heaters.
- The condensate is contaminated with traces of dissolved hydrocarbons, phenols, NH_3 , H_2S , etc.

12.5.1 Water Hammer

Steam or water hammer (more properly called hydraulic hammer) is one process plant phenomenon familiar to the general public. I well remember trying to warm up the steam system of a large amine plant in Texas City in 1980, and feeling more than hearing the crescendo of crashes due to steam hammer. The cause of steam hammer is illustrated in Fig. 12.7.

In general, *hydraulic hammer* is caused by the sudden conversion of the velocity of a liquid into pressure, causing a surge of pressure

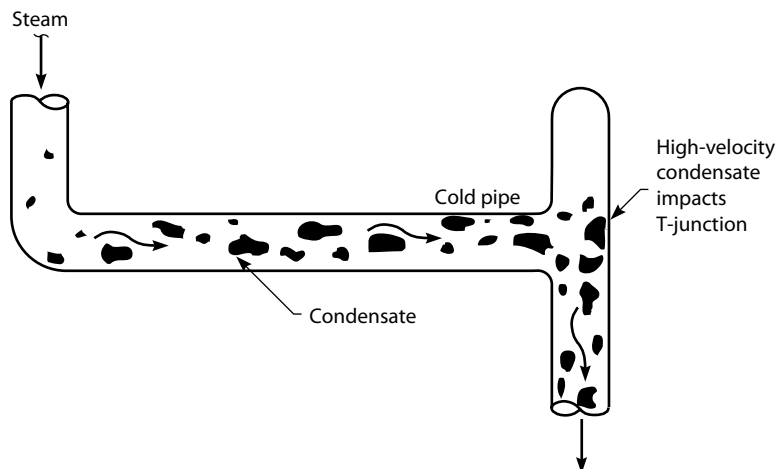


FIGURE 12.7 Steam hammer in piping.

inside a piping system. *Steam hammer* is caused by the creation of localized cool areas in piping systems containing saturated steam. For example, when first introduced into idled pipes, the steam condenses rapidly upon encountering a length of cold piping. This creates slugs of water. The continuing rapid but localized condensation of steam further downstream creates areas of low pressure or even a partial vacuum. The slugs of water rush to these areas of low pressure. Or, more precisely, the pressure differences created by the localized condensation of the steam provide a source of energy to accelerate water in the steam system. When these rapidly moving slugs of water impact an elbow or T-junction, the vibrations and noise of steam hammer result.

Introducing condensate from high-pressure steam traps into the low-pressure condensate collection system will generate steam. For example, passing 1000 lb/h of condensate from a 160-psig reboiler into a 20-psig collection system will generate about 100 lb/h of steam. More importantly, the volume of flow will increase from 20 ft³/h to over 1000 ft³/h, as a result of flash vaporization. If water at, say, 200°F from a low-pressure trap enters the piping, then any flashed steam will rapidly condense and create an area of low pressure. Slugs of water will rush to this low-pressure area, crashing into elbows and piping fittings; thus, the origin of steam hammer.

Operators stop steam hammer by dumping either the hot condensate (from a high-pressure steam trap) or the colder condensate (from a low-pressure steam trap) to the sewer. Either way, the condensate is no longer recycled to boiler feedwater. The sort of design required to collect condensate without steam hammer is illustrated in Fig. 12.8. Basically, hot, high-pressure condensate is collected in a

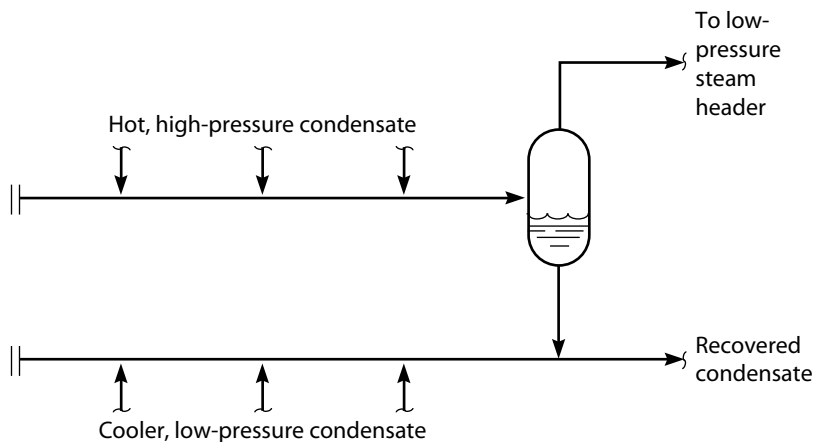


FIGURE 12.8 Avoiding water hammer in a condensate collection system.

dedicated system. The flashed steam from this system is recovered as low-pressure steam. The resulting water is then passed into the low-pressure condensate recovery piping system along with water flowing from the low-pressure steam traps.

I was told by an attendee at one of my seminars about the method used at his plant in Delhi to suppress steam hammer. A screen that looks like a pump suction filter is inserted between pipe flanges. As I have never used such a device myself and do not know how to calculate the required hole size or percent of open area of the screen, I have not included any further details in this text. My Indian colleague noted that this screen retrofit was quite effective in reducing steam hammer in this facility.

12.5.2 Condensate Backup in Reboilers

Operators who have problems with loss of reboiler capacity often attribute these problems to condensate backup. This is usually true. To drop the level of water out of the channel head, either the steam trap or the condensate drum is bypassed by putting the condensate to the sewer. Sometimes the float of the trap is sticking, but mostly the difficulty is an erratically high pressure in the condensate collection piping. Our guess is that the engineers who design these collection systems do not anticipate the large volume of steam generated from flashing, high-pressure condensate.

Condensate pumps are sometimes used to overcome such back-pressure problems. However, these pumps are often not kept in good repair, and condensate is still lost to the sewer. Eliminating the steam inlet control valve of the type shown in Fig. 12.4 has helped recover condensate from many reboilers supplied with low-pressure steam.

12.5.3 Contaminated Condensate Reuse

Much of the steam consumed in process units comes into direct contact with the process streams. A few examples are

- Steam vacuum jets
- Catalyst lift steam
- Stripping steam
- Vessel purge steam

For corrosion and safety reasons, the condensate recovered from these sources is best not returned to the deaerator for use as boiler feedwater.

However, there are no unchangeable rules in the process industry. Several years ago, I was working for an asphalt refinery in Corpus Christi, Texas. This refinery collected all their contaminated steam condensate streams from reflux drums and from a vacuum tower hot-well system. Without even steam stripping, the collected condensate was recycled back to the steam boiler. It's true only low-pressure (100-psig) steam was generated. It's true the steam so generated was not used in a turbine or in any chemical process. But on the other hand, they had been doing this for 20 years without any ill effects.

Had they asked my opinion, I would have told them based on 38 years of experience that I had never heard of any plant using contaminated water to generate steam. Luckily for them, they hadn't asked for my opinion.

Depending on the contaminant, the condensate may be reused for a number of services. Our favorite reuse of such contaminated condensate is as a replacement for *velocity steam* in the heater-tube passes of a fired furnace.

There is little danger in injecting a controlled amount of water into a furnace inlet when using a properly designed *metering pump*. Such pumps typically have a capacity of 1 to 10 GPM and provide a set flow regardless of the discharge pressure. The injected water flashes immediately to steam inside the furnace tubes. I have retrofitted several vacuum and delayed coker heaters with condensate injection systems with no adverse downstream effects. Water from the hot well of a vacuum ejector system is the normal source of condensate for this environmentally friendly modification.

12.6 Deaerators

The humble deaerator, operated by the utility department, is an interesting and important component of any process facility. Oxygen is a highly corrosive element and if left in the boiler feedwater would rapidly oxidize the boiler's tubes.

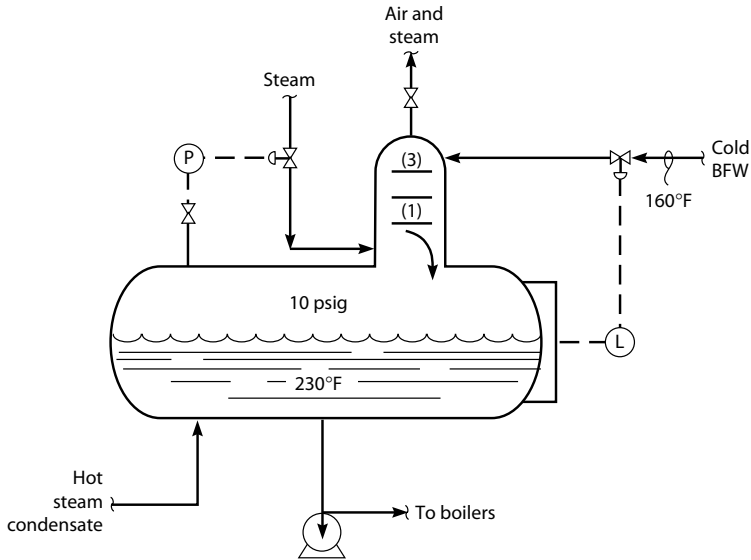


FIGURE 12.9 A deaerator.

The dissolved air left in boiler feedwater (BFW) is stripped out with steam in the deaerator shown in Fig. 12.9. The cold BFW has been taken first from the Mississippi River and then filtered to remove sand and sediment. Removal of the bulk of the calcium salts that would cause hardness deposits in the boilers is often accomplished by hot-lime softening. If excess CO_2 gas appears in downstream units consuming the steam, it is the fault of the lime softening, not the deaerator.

Figure 12.9 shows that the cold BFW has been heated to 160°F, from the 90°F lime-softener effluent. This is an excellent way to save steam to the deaerator. Usually, heating river or well water above 130°F would cause the laydown of hardness deposits inside the cooling-water tubes (assuming the cooling water to be on the tube side). But using softened water as a cooling-water supply permits that water to be easily heated to 160°F without fear of precipitating hardness deposits. Thus softened or demineralized cooling water can also be used as boiler feedwater, and about 40 percent of the deaerator steam requirements are then saved.

Most of the steam supply to a deaerator is used to heat the 160°F BFW to 230°F. This is the boiling point of water at 10 psig, which is the pressure in the deaerator. The water in the deaerator must always be at its boiling point; it is impossible to steam-strip air out of water below its boiling-point temperature.

The 160°F BFW is efficiently mixed with the incoming steam in what is effectively a small, vertical stripping tower mounted above the large deaerator drum. The majority of the steam condenses by

direct contact with the 160°F BFW and in so doing, the latent heat of condensation of the steam is used to increase the sensible-heat content of the 160°F BFW to 230°F.

A small amount of steam is vented from the top of the stripping tower to the atmosphere. Using a *gate valve*, with a hole drilled through the gate, is a simple way to control the venting rate. The dissolved air in the cold BFW is vented with this steam.

The stripped 230°F BFW drains into the large deaerator drum. This drum simply provides residence time for the high-pressure BFW pumps, which supply water directly to the boilers. Recovered steam condensate, which should be air-free, is fed to this drum through a separate nozzle.

12.6.1 Deaerator Flooding

One of the most interesting troubleshooting assignments of my career involved a deaerator in New Orleans. The operator reported loss of the water level in the deaerator drum. The level-control valve shown in Fig. 12.9 would open 100 percent, but this drove the level down even faster. The operator reported that the only way he could restore the water level would be to mostly close the water makeup level-control valve.

On the surface, this story sounds crazy. But let's see what happened. This deaerator had been designed for a much smaller flow of 160°F BFW and a much larger flow of hot-steam condensate than are current operations. The cold BFW feed line had been oversized, but the steam line was of marginal size. As the demand for hot BFW increased, the cold-BFW level-control valve opened. This reduced the temperature and pressure in the deaerator drum. In response, the steam pressure-control valve also opened. But when the cold-BFW level-control valve was 40 percent open, the steam pressure-control valve was 100 percent. Steam flow was now at its maximum.

The pressure inside the deaerator started to drop, as there was not enough steam flow to keep the water in the drum at its boiling point. The reduction in the deaerator pressure increased the volume of steam flow through the bottom tray of the stripping tower. Why?

According to the *ideal-gas law*:

$$\text{Volume} \sim \frac{1}{\text{pressure}}$$

This means volume is inversely proportional to pressure. As the pressure of the flowing steam declined, the steam's volume increased. The larger volume of steam flow resulted in a higher vapor velocity to the stripper trays 1 through 3. This caused the bottom stripper tray to flood. After a few minutes, the flooding forced cold BFW out of the atmospheric vent. The level in the deaerator drum then fell. The cold BFW level-control valve opened further, driving down both the temperature and the pressure in the deaerator drum. The volume and the

velocity of the steam flow to the stripper also increased, and the flooding became progressively more severe.

The only way the operator could restore the water level was to reverse this process. He manually restricted the flow of cold BFW through the level-control valve. This raised the stripper pressure and stopped the flooding.

The problem was corrected by increasing the temperature of the cold BFW to 190°F. Note that the key to solving this problem was observing the loss of water through the atmospheric vent. Sometimes, as far-fetched as an operator's description of a problem seems, it is still correct.

12.7 Surface Condensers

12.7.1 Steam Turbine Surface Condensers

Steam used to drive a turbine can be extracted at an intermediate pressure for further use of the low-pressure steam. Rarely is the steam vented to the atmosphere, as this wastes steam and the condensate is also lost. Many turbines exhaust steam under vacuum to a surface condenser. The lower the pressure in the surface condenser, the greater the amount of work that can be extracted from each pound of steam (see Chap. 24, Steam Turbines).

Discounting the presence of air leaks, the temperature inside the surface condenser determines the pressure of the steam exhausting from the turbine. This pressure is the vapor pressure of water at the surface condenser outlet temperature.

The original steam condensers were *barometric condensers*, which were used to increase the efficiency of the steam-driven reciprocating beam engines by a factor of 10. The barometric condenser was invented by James Watt (the steam engine was invented by Thomas Newcomen). Exhaust steam is mixed directly with cold water. As this creates a vacuum, the barometric condenser must be elevated about 30 ft above grade. The mixed condensate and cooling water drains through a pipe called a *barometric leg*—hence the name *barometric condenser*.

The *surface condenser* is an improvement on the barometric condenser, because it permits recovery of clean steam condensate. Other than this factor, the old-fashioned barometric condenser is more efficient than the more modern surface condenser.

Figure 12.10 shows the type of surface condenser most widely used on older steam turbines. Note that it has both a vapor and a liquid outlet. The turbine is located above the surface condenser. The wet exhaust steam flows down into the top of the condenser shell. Note that the exhaust steam from an efficient turbine will contain several percent of water.

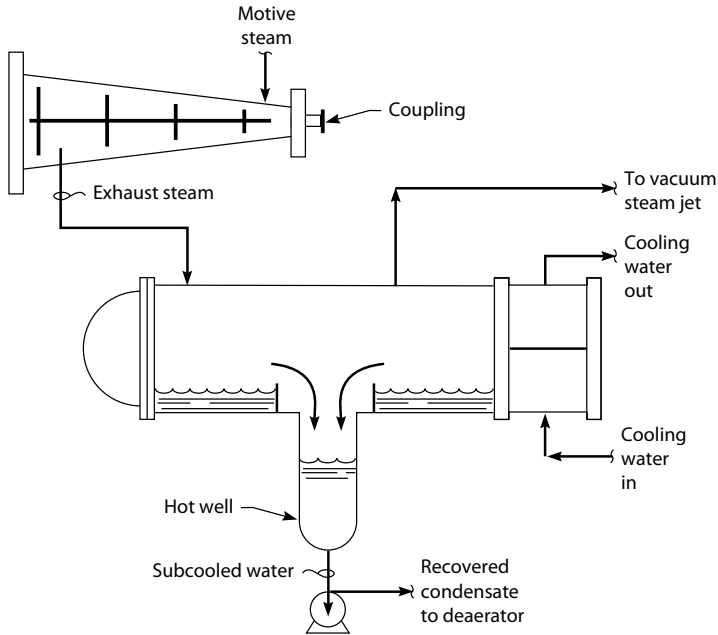


FIGURE 12.10 A surface condenser serving a steam turbine.

The shell-side pressure drop of the surface condenser is quite low. The vapor outlet flow consists of air drawn into the system through leaks, CO_2 , and a small amount of uncondensed steam. The weight flow of vapor from the top of the condenser is 1 percent or less than the flow of condensate from the bottom of the condenser.

The vapor is drawn into a steam jet (discussed in Chap. 23). The steam condensate flows into the boot or *hot well*. The water in the boot is slightly subcooled. This is accomplished by a pair of baffles that create a small zone of condensate backup. The subcooled condensate, cooled to perhaps 10°F below its boiling or bubbling point, is easier to pump. As the pressure in the hot well is sub-atmospheric, the hot-well pump typically develops a ΔP of at least 30 to 50 psi.

If the hot-well pump cannot handle the required condensate flow, the water level in the well will back up. The temperature in the hot well will go down as the lower condenser tubes are submerged. But this will reduce the surface area of the condenser that is exposed to the condensing steam. The condenser outlet temperature will therefore rise and so will the surface condenser pressure. This reduces the horsepower that can be recovered from each pound of the motive steam flowing to the turbine. The turbine will then slow down.

One common error made in monitoring the performance of surface condensers is the practice of considering the hot-well temperature as if

it were the true condensing temperature. It is the vapor outlet temperature that is the real surface condenser temperature. A decrease in the hot-well temperature resulting from a high hot-well water level is not an indication of improved condenser performance. It is a sign of reduced condenser capacity.

12.7.1.1 Air-Cooled Surface Condensers

Figure 12.11 shows a surface condenser elevated above the steam turbine. This creates an additional problem, in that moisture from the turbine exhaust steam will accumulate in the bottom of the turbine case. A special drain line from the turbine's case is needed to prevent condensate backup from damaging the spinning wheels.

One such turbine in a refinery near London would not drain properly. In order to push the condensate out of the turbine case, the operators were forced to raise the surface condenser pressure from 100 to 250 mm Hg (i.e., 20 inches of mercury vacuum, in the American system). Note that the balance line shown in Fig. 12.11 keeps the pressure in the turbine case and the condensate drum, into which the turbine case is draining, the same.

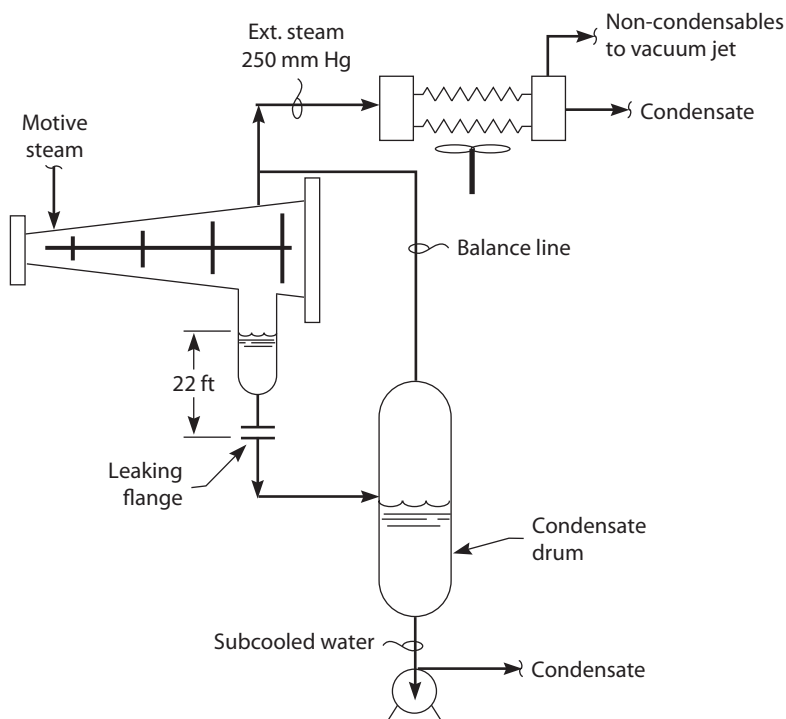


FIGURE 12.11 A leaking flange causes condensate backup.

The turbine case pressure was increased by raising the pressure in the air-cooled surface condenser. This was accomplished by shutting off several of the air fans, which, in turn, increased the condensing temperature of the exhaust steam. But why would raising the turbine case pressure drain the turbine anyway? After all, increasing the surface condenser pressure also increased the pressure in the drum that the turbine case drained to.

The answer is revealed when the available data are converted to consistent units. The 250 mm Hg turbine case pressure is equal to a height of water of 11 ft. Atmospheric pressure in London was 14.5 psia that day, which is equal to 33 ft of water. The difference between atmospheric pressure and the pressure in the turbine case, expressed in feet of water, is then

$$(33 \text{ ft} - 11 \text{ ft}) = 22 \text{ ft}$$

This turned out to be the exact elevation difference between the water level in the turbine case and the only flange on the drain line, from the turbine case to the drum below. This flange was found to be leaking. If the water head pressure at the flange became less than atmospheric pressure, then air was forced into the drain line. The bubbles of air expanded as they floated up the drain line into the turbine case. This prevented the water from draining freely through the drain line into the drum. Only by raising the turbine case pressure to pressure-up the flange to match atmospheric pressure could this air leak be stopped.

The flange leak was taped over, and the exhaust-steam pressure dropped back to 100 mm Hg. The steam required to drive the turbine fell by 18 percent. This incident is technically quite similar to losing the downcomer seal on a distillation tower tray. Again, it illustrates the sort of field observations one needs to combine with basic technical calculations. This is the optimum way to attack and solve process problems.

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CHAPTER 13

Vapor Lock and Exchanger Flooding in Steam Systems

The interaction of a steam reboiler with its condensate collection system can be very simple or very complex. Simple in the sense that the plant operators are satisfied to drain the steam condensate to the sewer from the channel head of the steam reboiler or feed preheater. Complex if the operators are trying to route the condensate back to the steam generation (boiler house) for recovery and reuse.

In many refineries, the ratio of fresh BFW (boiler feedwater) to steam generation is so high (70–80 percent) that one has to wonder, what has happened to most of the steam condensate? Mainly, due to design errors, the plant operators have been forced to divert the condensate to the sewer.

13.1 Function of the Steam Trap

The purpose of the steam trap, as shown in Fig. 13.1, is twofold:

1. To preserve the condensate seal. If steam blows through the outlet drain nozzle on the channel head, the rate of heat transfer will drop dramatically.
2. To minimize condensate backup. If condensate fills up tubes, then those tubes are unavailable to condense steam. Thus, the rate of heat transfer will drop in proportion to the tubes filled with water.

The steam trap is designed to pass liquids (H_2O) and retard the passage of vapor (steam).

There are two problems with the use of the control scheme shown in Fig. 13.1. First, steam traps are mechanically unreliable. Second, as valve A closes, the pressure in the lower half of the channel head may briefly fall below the frequently unstable pressure in the condensate

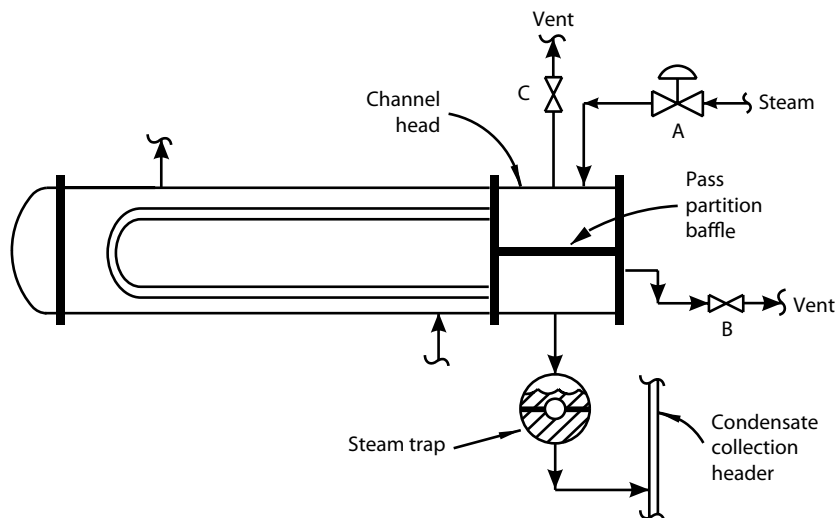


FIGURE 13.1 The function of a steam trap is to minimize condensate back-up and to preserve the condensate seal.

collection header pipe. When this happens, water in the condensate header could flow backwards into the channel head of the reboiler and rapidly submerge many of the tubes.

I would ignore this problem if the steam pressure is 100 psig or more and the condensate collection header is 30 psig or less. For smaller steam flows, 5000 to 10,000 lb/hr, I would use a steam trap. For larger steam flows, I would use a condensate pot and level control valve (LRC), as discussed in a following section.

The steam trap neither aids nor retards condensate drainage, unless it is mechanically malfunctioning, meaning it is sticking either open or closed. Also, replacing the steam trap with a condensate drum and LRC also neither retards nor aids drainage. Condensate drainage is a function of system hydraulics, as discussed in subsequent sections.

13.2 Non-condensable Venting

There is always a small amount of CO_2 in steam. It originates from residual carbonates in the BFW. Compared to steam, the CO_2 is a non-condensable with a limited solubility in H_2O . As the steam condenses, the few ppm of CO_2 will be trapped on the steam side of the tube bundle inside the channel head. However, none of the CO_2 will be trapped above the pass partition baffle shown in Fig. 13.1. And all of the CO_2 will be trapped below the pass partition baffle. Therefore, the problem is that, venting from valve **C** in Fig. 13.1 will vent off steam, but no CO_2 . To purge the accumulated CO_2 from the tube bundle, vent valve **B** must be used.

If this non-condensable is not removed from the channel head, then my clients have observed (over a period of days, not hours) a progressive loss in heat transfer capacity. The proof that this is a buildup of non-condensables is that briefly blowing the condensate seal restores heat transfer capacity. The CO_2 increases the tube-side heat transfer film resistance, rather similar to a temporary fouling deposit. I call this vapor binding.

In 1983, I was told by a Shell engineer that their standard is to continuously vent 0.5 percent of the steam through a restriction orifice to purge CO_2 . I still follow this practice, even though I suspect that just a small fraction of this rate of venting is required, if the steam is produced from demineralized water.

13.3 Corrosive Steam

If CO_2 is not vented off from valve **B**, then it will begin to dissolve in the condensate to form H_2CO_3 (carbonic acid). This acid causes tube failures due to acidic corrosion. Venting gas below the pass partition baffle will stop this corrosion, according to the Shell expert. I don't have any personal experience in stopping CO_2 corrosion with such venting. But it seems to make good engineering sense to me.

13.4 Condensate Drum

There is no process reason to use the condensate drum shown in Fig. 13.2 rather than a steam trap. In a modern refinery with large steam flows, the convention is to design systems with condensate

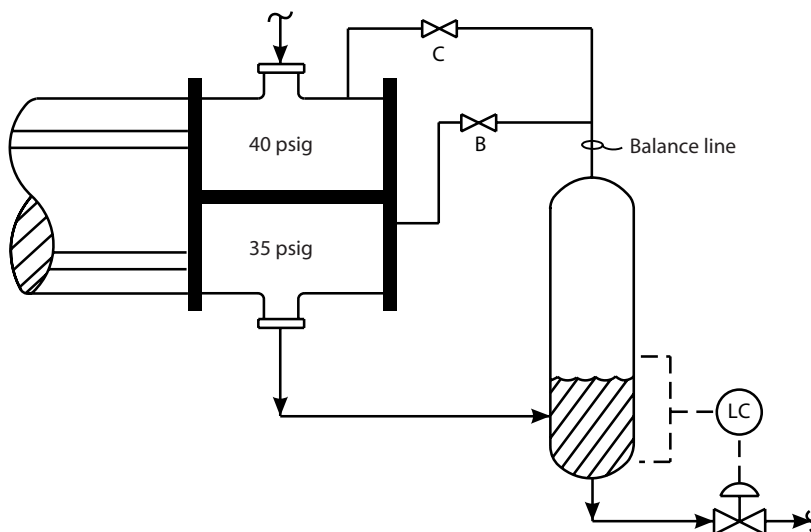


FIGURE 13.2 Valve B not C is the correct connection for the balance line.

drums for mechanical reliability. The balance line shown is required to maintain steam pressure in the condensate collection drum. The natural tendency is to maximize the drum pressure to promote condensate drainage from the drum by keeping valve **C** open and **B** closed. Then the pressure in the condensate drum would be 40 psig. But the pressure in the lower half on the channel head would only be 35 psig. (I've assumed a 5 psi ΔP on the steam side.) How can water at a pressure of 35 psig flow into a vessel at 40 psig?

The only possibility is condensate backup equivalent to 5 psi:

$$(5 \text{ psi}) \cdot (2.3 \text{ ft/psi}) = 11\frac{1}{2} \text{ ft}$$

Which is ridiculous, as the reboiler is only 5 ft in diameter. Even a 1 psi ΔP would submerge half the tubes with water. The operators will respond to this design error by draining the channel head directly to the sewer. Thus, the correct balance line connection location to use is through valve **B**, and never valve **C**. That is, leave **C** closed and **B** open. I learned about this from Sandy Lani.

13.5 Condensate Drainage and Vapor Lock

As steam condensate flows through process piping, it loses pressure due to frictional losses. Alternately, condensate draining from a reboiler outlet nozzle will accelerate and lose pressure according to the following formula:

$$\text{Pressure drop (psi)} = (0.015)(V^2)$$

where V = feet per second

For example, condensate draining through a nozzle at 10 ft per second would undergo a ΔP of 1½ psi. As the condensate is presumed to be draining out of the reboiler at its saturated liquid temperature (or boiling point or bubble point) as it loses the 1½ psi, you might think it would have a tendency to partially vaporize to steam. But this is not the case.

Let's say I have 100-psig steam condensing to water at its saturated liquid temperature of 320°F. If 1 cubic foot (about 59 lb) of this water is turned into steam, it will expand to approximately 150 cubic feet. The resulting increase in velocity will exponentially increase the pressure drop, which will largely choke off the flow. This is called *vapor lock*. To the operators, it appears as if a valve has been partially closed in the drain line. Depending on the line velocity and length, the flow will be reduced a lot or a little. For a short, oversized line, the effect may be minimal. But for a line sized for flowing water only, at a design velocity of 8 or 12 ft per second for a length of several hundred feet, the effect will be extremely large.

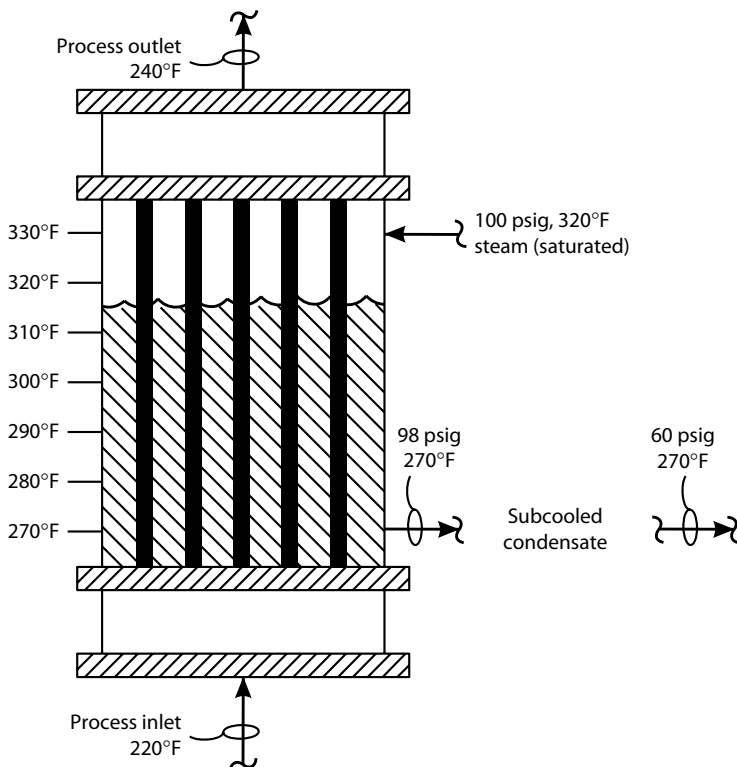


FIGURE 13.3 The effect of vapor lock is to promote condensate backup and subcooling of the condensate.

13.5.1 Effect of Vapor Lock

The consequences of vapor lock on the performance of a vertical thermo-siphon steam reboiler, with steam on the shell side, is shown in Fig. 13.3.

The boiling point temperature of 60-psig water is about 270°F. The condensation temperature of 100-psig steam is about 320°F. As the condensate flows out of the shell side of the reboiler, it loses:

- Two psi through the nozzle.
- Plus 38 psi through the condensate drain line.

To suppress the evolution of steam in the drain line, the condensate must be subcooled from 320°F to 270°F as it leaves the reboiler. This can only be accomplished by condensate backup. And if the condensate is not subcooled to 270°F, then the condensate will begin to vaporize. Which will then:

- A. Choke off the flow.
- B. Back up the shell-side water level.

- C. Cause the condensate to be subcooled, so that when it falls to 60 psig, it is cold enough so that it does not flash and vapor lock is suppressed.

The problem with condensate backup and subcooling is the very low heat transfer coefficient (U) for subcooling vs. condensation. Typical heat transfer coefficients ($\text{Btu/hr/ft}^2/^{\circ}\text{F}$) are:

- Steam condensation = 400 to 600
- Subcooling = 20 to 40

To condense a pound of 100 psig steam requires 920 Btu of heat removal. To subcool water from 320°F to 270°F requires 50 Btu of heat removal. The reader can conclude from the above that around half of the surface area of the reboiler in Fig. 13.3 is devoted to subcooling, but only for 5 percent of the reboiler duty [i.e., $50 \div (920 + 50) = 5$ percent].

13.5.2 Design Options to Mitigate Vapor Lock

- Option One—Double the surface area of the reboiler.
- Option Two—Increase the size of the condensate drain line by an order of magnitude. I have actually done this and it worked. However, in general, this is not a cost-effective option for longer condensate collection drain lines.
- Option Three—Inject cold water into the condensate run-down to suppress vaporization. To suppress vaporization of 320°F condensate, as shown in Fig. 13.3, enough 70°F water (i.e., 25 percent of the condensate flow rate) would have to be injected. I have done this as well, with good results.
- Option Four—Drain condensate to the sewer beneath the reboiler. This unfortunate option is, by far, the most usual solution.
- Option Five—Install a condensate pump and condensate drum. Pump the condensate out under level control. The drum must be immediately next to and underneath the reboiler. Also, the drum must be high enough above the new condensate pump to provide sufficient net positive suction head (NPSH) for the pump. This is, of course, the preferred, if not the typical way of handling this very common design problem.

13.6 Elevated Condensate Collection Drum

Figure 13.4 illustrates a condensate drum that is elevated above the reboiler steam trap. For simplicity, let's assume that 15 psi of pressure (i.e., 1 atmosphere) equates to 35 ft of hot water head pressure.

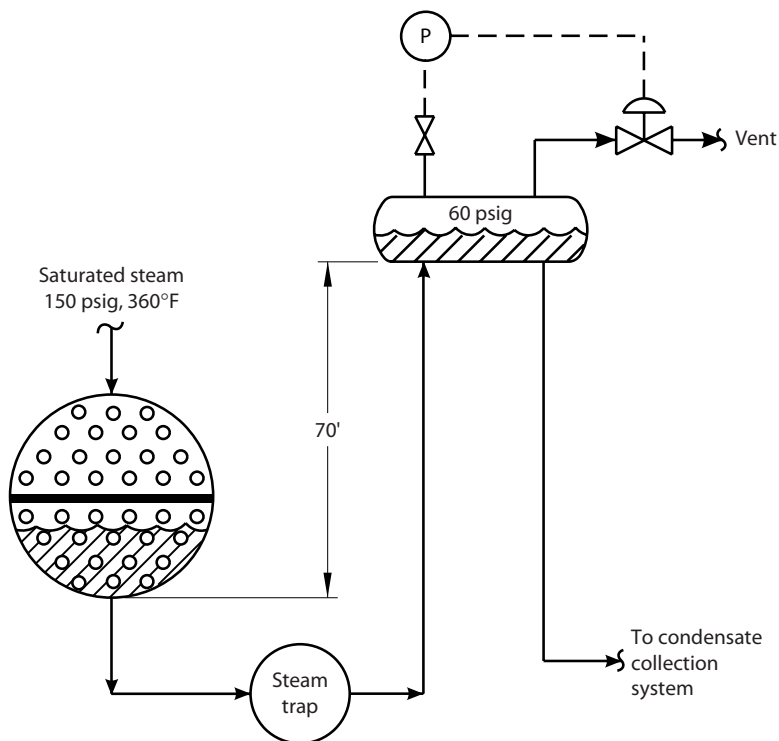


FIGURE 13.4 Elevating a condensate drum above the reboiler, regardless of the steam supply pressure, will result in loss of reboiler capacity.

Note that the steam supply pressure is 150 psig with a saturation temperature of 360°F.

Assume that the condensate drains from the reboiler as saturated water (150 psig and 360°F). The ΔP through the steam trap is zero. The hot water will lose 70 ft or 30 psig of pressure as it rises up to the drum, which operates at only 60 psig. The excess ΔP above the minimum required is then:

$$150 - 30 - 60 = 60 \text{ psi}$$

Thus, superficially, it appears as if the condensate could easily flow up into the drum. But it will not.

The problem is that the condensate rising from the steam trap will begin to lose pressure due to the gain in elevation. As it rises above the reboiler level, the hot saturated water will start to flash, undergo vapor lock as described above, and thus choke off the flow. The condensate will back up in the channel head of the reboiler shown in Fig. 13.4 and become subcooled enough to suppress steam evolution until the condensate enters the drum. The required subcooling may easily flood half or more of the surface area of the reboiler. The loss of reboiler

capacity typically will not be tolerated by the plant operators, who will then be forced to drain the valuable steam condensate to the sewer.

I once had this problem on a debutanizer reboiler and decided to add a parallel steam trap. Needless to say, it did not help at all. But everyone has to learn from their own mistakes. Injecting cold water at the base of the riser did alleviate the problem.

13.7 Conclusion

The design of a steam condensate collection system is a complex task. It cannot be approached just as a problem in hydraulics, because it is also a problem in vapor-liquid equilibrium and in the conversion of sensible heat to latent heat. Considering these three factors together requires quite complex calculations, for which I have had to rely on the use of a computer model. These same sorts of considerations apply to multicomponent hydrocarbon systems, such as propane-butane refrigeration loops, but then the calculation complexity increases by an order of magnitude.

CHAPTER 14

Bubble Point and Dew Point

Equilibrium Concepts in Vapor-Liquid Mixtures

Our work as process engineers and operators is based on three principles:

- *Hydraulics*—when the velocity of water in a pipe goes up, its pressure goes down.
- *Heat balance*—condensing steam gives up latent heat, or sensible heat, to increase the temperature in the feed preheater.
- *Vapor-liquid equilibrium*—a boiling liquid is at its bubble point, and a condensing vapor is at its dew point.

14.1 Bubble Point

The purpose of this chapter is to explain what is meant by the terms *bubble point* and *dew point*, and how we can use these ideas to improve the operation of the distillation tower. To begin, we will derive the *bubble-point equation* from the basic statement of vapor-liquid equilibrium:

$$y_1 = K_1 \times x_1 \quad (14.1)$$

where y_1 = concentration of the first component in the vapor
 x_1 = concentration of the first component in the liquid
 K_1 = an equilibrium constant

We really should use *mole fraction*, and not concentration, in our description of y and x , but for our work, we will just say that the term *concentration* refers to the percent of a component that the operator would see in the gas-chromatographic (GC) results, as reported by

the lab. The equilibrium constant, assuming the *ideal-gas law* applies, is defined as

$$K_1 = \frac{P_{V,1}}{P_T} \quad (14.2)$$

where $P_{V,1}$ = vapor pressure of the first component, at the temperature we are working at, in psia (see Fig. 14.1 for a chart of vapor pressures used here)

P_T = total pressure in psia (psia = psig + 14.7)

If you do not recall the meanings of *mole fraction* or the *ideal-gas law*, don't worry—it is not necessary to recall these in order to understand bubble points, or dew points. Substituting Eq. (14.2) into Eq. (14.1), we obtain

$$y_1 = \frac{P_{V,1}x_1}{P_T} \quad (14.3)$$

Let's assume that we have three components in the vessel shown in Fig. 14.2. Then we could write

$$y_1 + y_2 + y_3 = \frac{P_{V,1}x_1}{P_T} + \frac{P_{V,2}x_2}{P_T} + \frac{P_{V,3}x_3}{P_T} \quad (14.4)$$

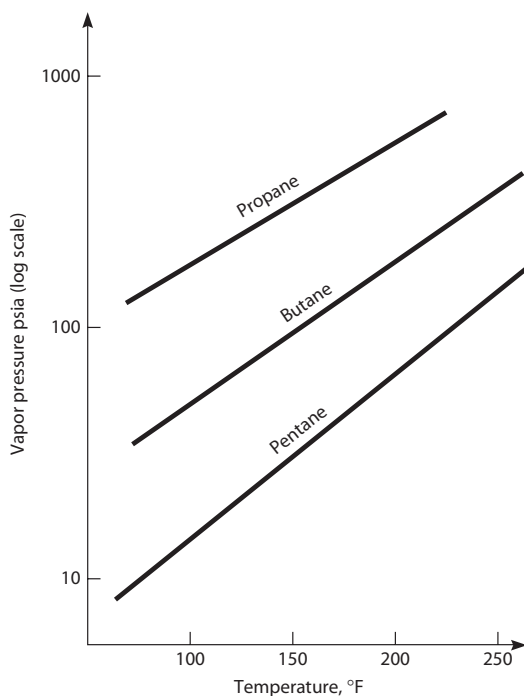
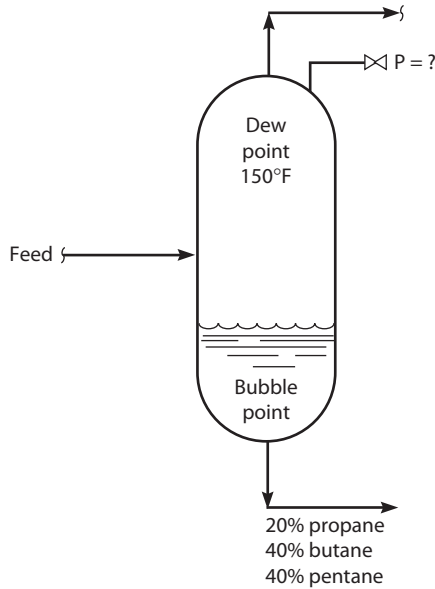


FIGURE 14.1 Vapor pressure chart.

FIGURE 14.2

Calculating bubble-point pressure.



But if we add up the concentration of the three components in the vapor phase on the left side of Eq. (14.4), we would get 100 percent. The fractions on the right side of Eq. (14.4) all have the same denominator (i.e., P_T), so they can be also added together:

$$100\% = \frac{P_{V,1}x_1 + P_{V,2}x_2 + P_{V,3}x_3}{P_T} \quad (14.5)$$

Recalling that 100 percent of anything is the whole thing or, in other words, equals unity or one, if we cross-multiply both sides of this equation by P_T , we have

$$P_T = P_{V,1}x_1 + P_{V,2}x_2 + P_{V,3}x_3 \quad (14.6)$$

This is the bubble-point equation for a three-component system.

14.1.1 Using the Bubble-Point Equation

Are we missing the pressure P_T in the flash drum shown in Fig. 14.2? Let's calculate this pressure, using the bubble-point equation and the vapor pressure chart shown in Fig. 14.1:

Component	Vapor Pressure at 150°F, Psia		Concentrating Mol% in Liquid, %		Partial Pressure, Psia
Propane	330	×	20	=	66
Butane	105	×	40	=	42
Pentane	35	×	40	=	14
Vessel pressure				=	122

The term *partial pressure*, meaning part of the total pressure created by each component, is important. The partial pressure of a component divided by the total pressure is the concentration of the component in the vapor phase. For example, the concentration of propane in the vapor leaving the drum shown in Fig. 14.2 is

$$\frac{66 \text{ psia}}{122 \text{ psia}} = 54\%$$

What is the concentration of pentane in the vapor? (Answer: $14 \div 122 = 11.5$ percent.)

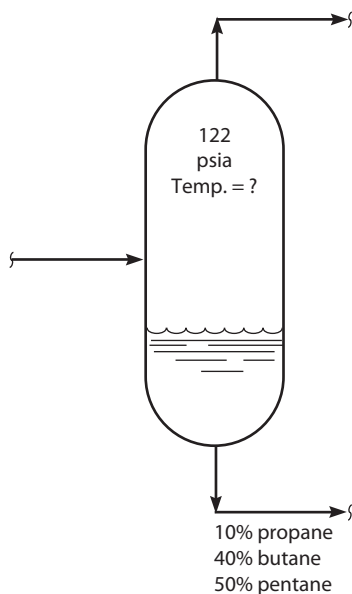
14.1.2 Adjusting Temperature to Meet a Product Specification

Let us assume that a new set of product specifications has just been issued to your shift. The liquid from the flash drum shown in Fig. 14.3 has too much propane. The new liquid specification is

- Propane: 10 percent
- Butane: 40 percent
- Pentane: 50 percent

The pressure in the drum is still fixed at 122 psia. So, it seems as if we will have to run the drum hotter. But how *much* hotter?

FIGURE 14.3
Calculating a temperature to meet a new specification.



Suppose we raise the drum temperature to 160°F and repeat our bubble-point calculation:

Component	Vapor Pressure at 160°F, Psia		Concentrating Mol% in Liquid, %		Partial Pressure, Psia
Propane	380	×	10	=	38
Butane	130	×	40	=	52
Pentane	40	×	50	=	20
Calculated vessel pressure				=	110

Apparently, our guess of 160°F was wrong. If we had guessed the correct temperature, the calculated vessel pressure would have been 122 psia, not the 110 psia. Try to work this problem yourself by guessing a new flash drum temperature (answer: 168°F).

This seems to be a potentially good application for computer technology. For example, an operator is running a debutanizer and finds that she has too much isobutane in her isopentane bottoms product. She enters the most recent gas chrome result from the lab in the computer, with the corresponding tower pressure and reboiler outlet temperature. Next, she enters the isobutane specification she would like to achieve in the tower's bottoms product. The computer then tells her to raise the reboiler outlet temperature by 17°F to get back on spec (specification) quickly. This is a lot better than guessing at the reboiler temperature, lining out the tower, and waiting half the shift for the lab GC result before making your next move.

14.2 Dew Point

14.2.1 Dew-Point Calculations

We will now derive the *dew-point equation* from the same basic statement of vapor-liquid equilibrium, starting with Eq. (14.3) in the previous section:

$$y_1 = \frac{P_{v,1}x_1}{P_T} \quad (14.3)$$

Now let's multiply both sides of this equation by $P_T/P_{v,1}$

$$\frac{y_1 P_T}{P_{v,1}} = x_1$$

Again, let's assume we have three components:

$$\frac{y_1 P_T}{P_{v,1}} + \frac{y_2 P_T}{P_{v,2}} + \frac{y_3 P_T}{P_{v,3}} = x_1 + x_2 + x_3 \quad (14.7)$$

However, if we add up the concentration of the three components in the liquid phase on the right-hand side of Eq. (14.7), we would get 100 percent, which is unity or equal to one as before:

$$\frac{y_1 P_T}{P_{V,1}} + \frac{y_2 P_T}{P_{V,2}} + \frac{y_3 P_T}{P_{V,3}} = 1.0 \quad (14.8)$$

Next, we divide both sides of the equation by P_T , the vessel pressure:

$$\frac{y_1}{P_{V,1}} + \frac{y_2}{P_{V,2}} + \frac{y_3}{P_{V,3}} = \frac{1}{P_T} \quad (14.9)$$

This is the dew-point equation for a three-component system.

14.2.2 Using the Dew-Point Equation

Is science really this easy? Much of the science applied to process engineering is straightforward. In order to show you how to calculate the temperature of the vapor leaving the depropanizer in Fig. 14.4, we will use the dew-point equation.

This time, we know that the tower-top pressure is 175 psig, or 190 psia. We also know that the composition of the overhead vapor is

- Propane: 80 percent
- Butane: 15 percent
- Pentane: 5 percent

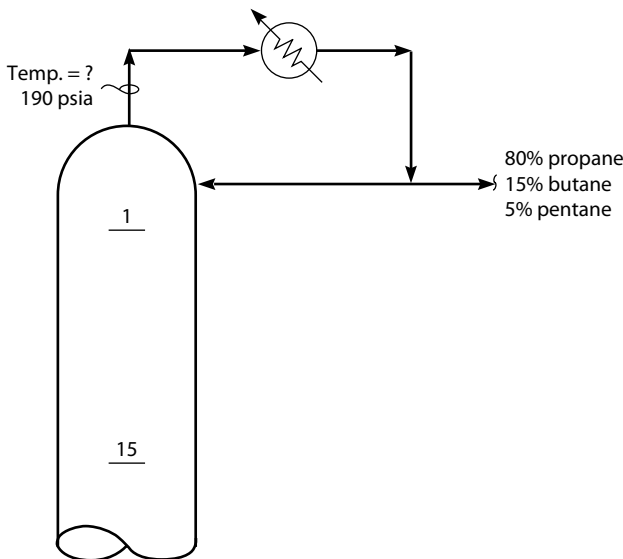


FIGURE 14.4 Calculating a dew-point temperature.

It is normal to assume that the vapor leaving the top of a tower is at its dew point. That is, it is at equilibrium with the liquid on the top tray of the tower. Unfortunately, this assumption falls apart if the tower is flooding and liquid is being entrained overhead from the column with the vapor. However, assuming a normal, nonflooded condition, we will guess that the tower-top temperature is 140°F. Using the vapor-pressure curves provided in Fig. 14.1, we would calculate as follows:

Component	Vapor pressure at 140°F, Psia, P_v	Concentration in Vapor y	$y \div P_v$
Propane	300	0.80	0.00267
Butane	90	0.15	0.00166
Pentane	30	0.05	0.00167
Sum of quotients $y \div P_v$			0.00600

According to Eq. (14.9), the sum of the quotient of $y \div P_v$ ought to equal $1/P_T$:

$$0.00600 = \frac{1}{P_T}$$

Solving for P_T , we find the calculated tower-top pressure equals 167 psia. But the actual tower-top pressure is 190 psia. Evidently, we guessed too low a temperature. Try the calculation again yourself with a better guess for the tower-top temperature (answer: 146°F).

For this tower, the composition of the overhead vapor is the same as the overhead liquid product made from the reflux accumulator.

14.2.3 Revised Product Specification

It seems that the depropanizer overhead composition specification has been changed. Our new operating orders are to produce

- Propane: 90 percent
- Butane: 8 percent
- Pentane: 2 percent

The tower-top pressure is still 190 psia. The tower-top temperature will have to be reduced. Let's guess that it will be reduced to 130°F:

Component	Vapor Pressure at 130°F, Psia	Concentration in Vapor y	$y \div P_v$
Propane	270	0.90	0.00333
Butane	80	0.08	0.00100
Pentane	26	0.02	0.00077
Sum of quotients $y \div P_v$			0.00510

Referring again to Eq. (14.9), we can solve for P_T , the tower pressure:

$$0.00510 = \frac{1}{P_T}$$

or $P_T = 196$ psia. But this is the calculated tower pressure. The actual tower pressure is only 190 psia. Try to repeat this calculation to get the correct tower-top temperature (answer: 128°F).

Again, this seems to be a rather nice application for computer technology. Even a good-quality programmable calculator can store a number of vapor-pressure curves. At least for hydrocarbons, equations for these curves can be extracted from the API (American Petroleum Institute) data book. Also, a programmable calculator can perform bubble-point and dew-point calculations, with over 10 components, without difficulty.

Dear reader, if you have skipped most of this chapter because of the equations and the math, please consider the following:

- Most graduate chemical engineers cannot do these calculations without the aid of a computer.
- Imagine the pleasure you will get if you're an operator explaining how to execute bubble-point and dew-point calculations to your unit engineer. Possibly you might observe, "Didn't they teach you this in your first year of chemical engineering at the university?"
- If you are a chemical engineer, you will have to admit that we all learned the rules of vapor-liquid equilibrium during our freshman year.

For me, it's easy to remember how to do bubble-point and dew-point calculations. I carry a pocket-size vapor pressure chart in my wallet. When I'm out in the plant troubleshooting, that's all I need to check temperatures versus pressures for light hydrocarbon systems.

Reference

1. American Petroleum Institute, *API Technical Data Book*, vol. I, sec. 5, "Vapor Pressures," Aug. 1964, fig. 5A 1.1, p. 5-3.

CHAPTER 15

Steam Strippers

Source of Latent Heat of Vaporization

The use of steam to remove lower-boiling or lighter components from a liquid is one of the oldest methods of distillation. Sometimes called *steam distillation*, this technique relies on a combination of two simple effects.

15.1 Heat of Evaporation

The first effect is illustrated when we blow across a bowl of hot soup to cool the soup. Our breath displaces the steam vapors that are on top of the soup. This encourages more molecules of steam vapors to escape from the soup; that is, the vapor pressure of the steam above the liquid soup is diminished, because steam is pushed out of the soup bowl with air. The correct technical way to express this idea is to say, "The partial pressure of the steam, in equilibrium with the soup, is diminished."

But our breath itself does not remove heat from the soup. The evaporation of steam from the soup, promoted by our breath, takes heat. Converting one pound of soup to one pound of steam requires 1000 Btu. This heat of evaporation comes not from our breath, but from the soup itself. The correct technical way to express this second effect is, "The sensible-heat content of the soup is converted to latent heat of evaporation."

15.1.1 Example Calculations

For example, if we have 101 lb of soup in a rather large bowl, and cause one pound to evaporate by blowing across the bowl, the soup will lose 1000 Btu. This heat of evaporation will come at the expense of the temperature of the remaining soup in the bowl; that is, each pound of soup will lose 10 Btu. If the specific heat of our soup is $1.0 \text{ Btu}/[(\text{lb})(^\circ\text{F})]$, the soup will cool off by 10°F .

A steam stripper, as shown in Fig. 15.1, works in the same way. The diesel-oil product drawn from the fractionator column is contaminated

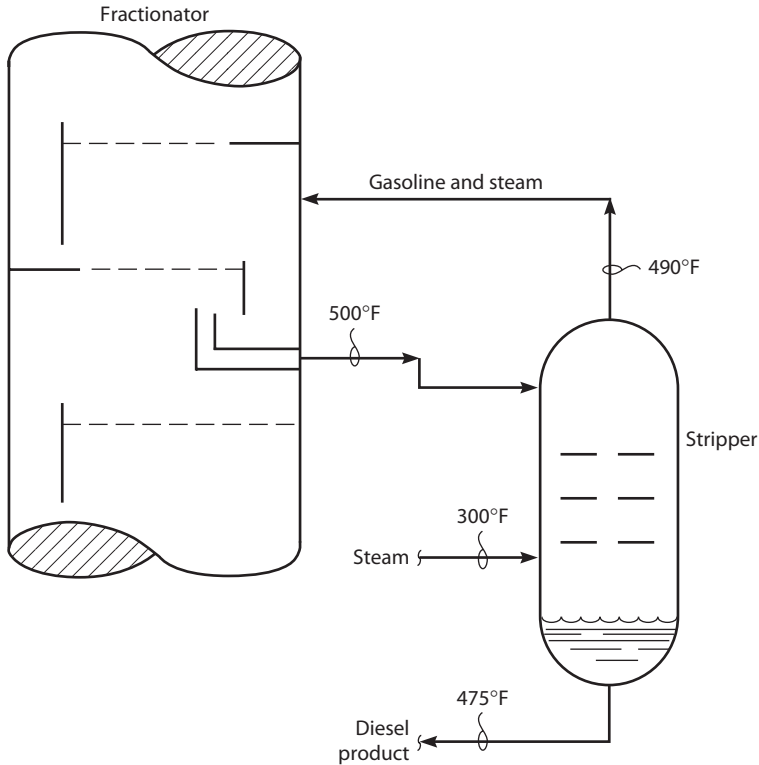


FIGURE 15.1 Heat of evaporation comes from product, not from steam.

with gasoline. The stripping steam mixes with the diesel-oil product on the trays inside the stripper tower. The steam reduces the hydrocarbon partial pressure and thus allows more gasoline to vaporize and to escape from the liquid phase into the vapor phase. The heat of vaporization of the gasoline cannot come from the steam, because the steam (at 300°F) is colder than the diesel oil (at 500°F). The heat of vaporization must come from the diesel-oil product itself.

We can use this idea to calculate the percent of diesel oil that would actually vaporize across the stripping trays in the stripping tower. Let's assume the following thermal properties for a typical hydrocarbon mixture of diesel and gasoline:

- 0.60 Btu/[(lb)(°F)] is the specific heat.
- 100 Btu/lb is the latent heat.

Referring to Fig. 15.1, the reduction in sensible heat of the diesel product equals

$$(500^{\circ}\text{F} - 475^{\circ}\text{F}) \times 0.60 = 15 \text{ Btu/lb}$$

The percent of the feed to the stripper that evaporates is then

$$(15 \text{ Btu/lb}) \div 100 \text{ Btu/lb} = 15\%$$

15.1.2 Measuring Evaporation in the Field

Note that we have neglected the heat picked up by the steam in the preceding calculation. Often, the steam flow is quite small compared to the stripper feed, so usually this effect may be disregarded. Unfortunately, we seldom should neglect ambient-heat losses.

Certainly, if the stripper tower and associated piping are radiating heat from the product, this is not contributing to stripping. To determine the temperature drop due to ambient-heat losses, proceed as follows:

1. Check the temperature of the stripper feed.
2. Shut off the stripping steam and wait 20 min.
3. Check the temperature of the stripper bottoms product.

The difference between the two temperatures represents the ambient-heat loss associated with the stripper. Of course, with no ambient-heat loss, this temperature difference would be zero.

15.2 Stripper Efficiency

Many side-stream steam strippers of the type shown in Fig. 15.1 do not work very well. Operating personnel report that the stripping steam is not effective in removing undesirable lighter components from the stripper feed. Why could this be so?

One of the main reasons for this sort of poor stripping efficiency is subcooled liquid feed to the stripper. Liquid drawn from any tower or vessel is assumed to be in equilibrium with the vapor phase in the tower or vessel. We say that the liquid is at its bubble point or boiling point. We say that the vapor in the vessel is at its dew point or saturation temperature.

When steam is mixed with a liquid at its bubble point, the partial pressure of the vapor in contact with the liquid is reduced. The liquid then begins to boil. The lighter components of the liquid are turned into vapor and are carried out of the stripper with the steam.

If liquid drawn from a column cools below its bubble point as a result of ambient-heat loss, we say it is *subcooled*. Mixing a small amount of steam with subcooled liquid will reduce the partial pressure of any vapor in contact with the liquid, but not enough to promote boiling. Eventually, as more and more steam is mixed with a subcooled liquid, it will begin to boil. But for a given amount of steam, the amount of vapor that can be boiled out of a liquid will

always be less if the liquid is subcooled. In this way, ambient-heat loss reduces the stripping efficiency of steam.

Wet steam will also reduce stripping efficiency. The water in the steam will be turned into steam when it contacts the hot diesel oil in the stripper, shown in Fig. 15.1. The heat of vaporization for this water must come from the sensible heat of the diesel. This reduces the temperature of the diesel, which also reduces its vapor pressure, which then makes it more difficult to vaporize its lighter gasoline components.

I was once working in a refinery that could not meet the flash-point specification for its diesel product. *Flash point* is the temperature at which a hydrocarbon will ignite when exposed to an open flame. To raise the flash point of diesel oil, it is steam-stripped to remove the lighter, more combustible components. I noticed that I could drain water from the bottom of the steam supply line to the diesel-oil stripper. I then screwed a *steam trap* onto the 0.75-inch drain valve on the steam supply line. The stripper bottoms temperature increased by 35°F, and the flash temperature of the diesel product increased from 120 to 175°F.

High liquid levels in the bottom of the stripper will also reduce stripping efficiency. A liquid level above the steam inlet will cause the stripping trays to flood. Flooding vastly decreases tray efficiency and hence stripping efficiency.

As discussed in Chap. 4, tray deck dumping also greatly reduces tray efficiency. Unfortunately, steam strippers can have widely varying vapor rates between the top and bottom trays of a column.

15.2.1 Vapor Distribution in Steam Strippers

Figure 15.2 shows the type of hydrocarbon stripper discussed above. Note that the vapor load to the bottom tray is only the 3000 lb/h of stripping steam. The vapor load from the top tray is 15,000 lb/h. In other words, the vapor leaving the top tray of the stripper consists of 3000 lb/h of stripping steam *plus* 12,000 lb/h of hydrocarbon vapor.

From the designer's point of view, the top tray of the stripper should have several times more sieve holes or valve caps on its tray deck than the bottom tray. If, however, all the trays in the stripper are identical, then either the bottom tray will leak (see Chap. 4) or the top tray will flood. Either way, stripping efficiency will suffer.

15.2.2 Steam Stripping Water

So far, we have been discussing the stripping of hydrocarbons. But of equal importance is steam stripping of aqueous streams such as

- Removing benzene from seawater
- Removing ammonia from sour water
- Removing hydrogen sulfide from amine solutions

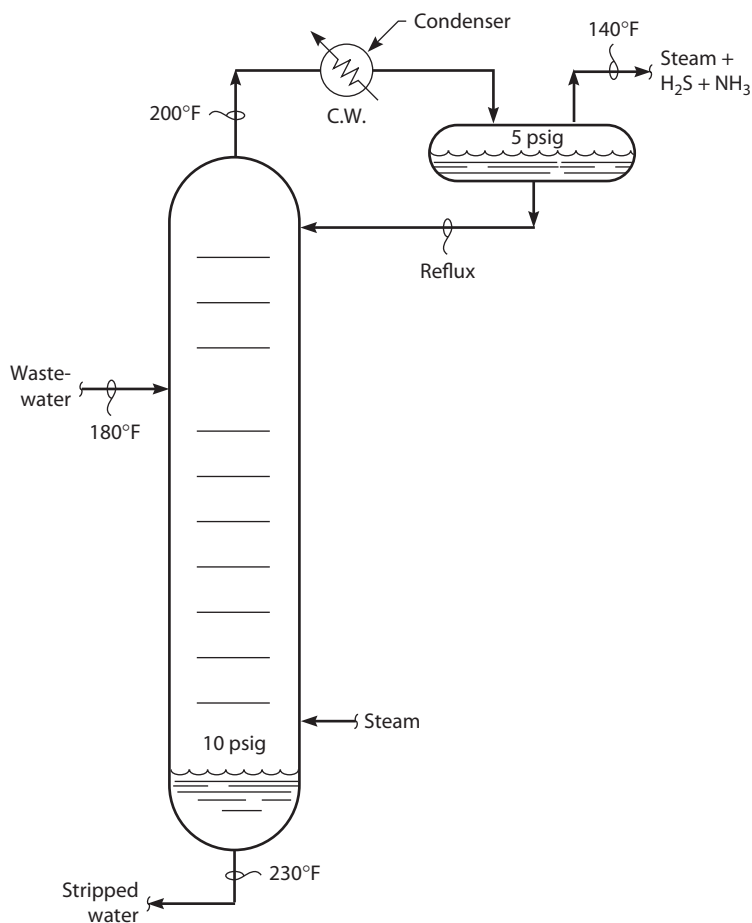


FIGURE 15.3 Steam stripping sour water.

was evolved or released. This is called the *heat of solution*. When these same gases are driven out of the wastewater with the stripping steam, this same heat of solution has to be supplied. Again, this heat comes from condensing the stripping steam in the water flowing across the trays in the tower.

3. Some of the stripping steam condenses in the overhead condenser, shown in Fig. 15.3. The condensed steam, which accumulates in the reflux drum, is totally refluxed back to the top tray of the stripper tower.
4. A small amount of the stripping steam remains as steam and leaves the reflux drum with the H₂S and NH₃ vapor product.

From this distribution of the stripping steam, we can conclude that the pounds of vapor to the bottom tray are much larger than the pounds of vapor leaving the top tray. This is just the opposite of the diesel-oil stripper.

15.2.3 Temperature Distribution in Water Strippers

The 230°F stripper bottom temperature, shown in Fig. 15.3, is simply the boiling point of water at the 10-psig tower-bottom pressure. Small amounts of extraneous chemicals (phenol, alcohols, aromatics) dissolved in the stripped water do not change the boiling-point temperature of this water.

The 200°F stripper tower-top temperature is the dew point of the vapors leaving the top tray. Most of these vapors are steam, and that is why the tower-top temperature is so high. The high steam content of the overhead vapors cause a water stripper to behave in a strange way! When the top reflux rate is increased, the tower-top temperature goes up, not down. This odd behavior is easily understood if we note that there is no liquid product made from the reflux drum. Therefore, the only way to increase the reflux rate without losing the level in the reflux drum is to increase the steam rate to the bottom of the stripper. The extra stripping steam drives up the tower-top temperature.

The 140°F reflux drum temperature, shown in Fig. 15.3, is the dew point of the vapors leaving the reflux drum. Almost always, we would like to minimize this particular temperature. The lower the reflux drum temperature, the smaller the amount of steam in the off-gas. If the off-gas is H_2S and NH_3 flowing to a sulfur recovery plant, the steam carried into the sulfur plant reduces the sulfur plant's capacity and efficiency.

On the other hand, a low reflux drum temperature increases the solubility of H_2S and NH_3 in the reflux water. As the concentration of H_2S and NH_3 in the reflux increases, the stripper has to work harder to keep these components out of the stripped water.¹

15.2.4 Reboiled Water Strippers

Many water strippers are initially designed with steam reboilers, rather than with open stripping steam. The amount of steam required is the same in either case. The great advantage of the reboiled stripper is that the steam condensate is recovered and recycled back to the boilers. When open stripping steam is used, the steam condensate is added to the stripped water, thus increasing the plant's water effluent. Hence, the use of open stripping steam is environmentally unfriendly.

Fouling is the main reason for abandoning many stripper reboilers and substituting open stripping steam. It is easy to pipe up steam

to the stripper. It is difficult to determine and control the cause of the fouling in the reboiler. But not to find and control the cause of the fouling is sloppy engineering and poor operation.

15.2.5 Stripping Aromatics from Wastewater

Removing benzene and other aromatic compounds from a plant's effluent water is an increasingly common environmental requirement. This is typically achieved with a steam stripper. There is a rather neat trick, which can increase the stripper's efficiency: adding saltwater to the stripper feed. Aromatics, especially benzene, are far less soluble in brine than they are in freshwater. But, of course, the brine will be more corrosive than salt-free freshwater.²

15.2.6 Side-Stream Stripper Hydraulics

One of my first successful projects as a young engineer was to improve the quality of jet fuel from a side-stream stripper. Initially, I had thought the problem was due to poor stripping efficiency, but it was really a simple problem in *hydraulics*: the part of technology that deals with the pressure drop of gases and liquids flowing through pipes and other process equipment. Liquid heads, gravity, and friction are all parts of hydraulics.

The particular problem I encountered is illustrated in Fig. 15.4. The jet fuel product was steam-stripped to remove a lighter naphtha contaminant. But much naphtha was left in the jet fuel. Apparently, the packing in the stripper tower was not working properly. However, a discussion with the unit operator indicated that they were using very little stripping steam. Introduction of a normal amount of steam resulted in a loss of liquid level in the bottom of the stripper.

The pressure at point A in Fig. 15.4 was 13 psig. This means that the pressure drop in the vapor line from the stripper back to the fractionator was 3 psig. In order for the unstripped jet fuel to flow out of the lower-pressure fractionator and into the higher-pressure stripper, it had to overcome this 3-psig pressure difference. The 16-ft elevation difference between the draw-off nozzle on the fractionator and the stripper inlet provided the necessary liquid head driving force.

Let us assume that the specific gravity of the unstripped jet fuel was 0.59. Also, note that for water, which has a specific gravity of 1.0:

$$1.0 \text{ psi of pressure} = 2.31 \text{ ft of water}$$

This means that the pressure head of a column of water 2.31 ft high equals 1 psig. The height of a column of unstripped jet fuel equal to a pressure head of 1 psig is then

$$1.0 \text{ psi} = 2.31 \times \frac{1.0}{0.59} = 4.0 \text{ ft of jet fuel}$$

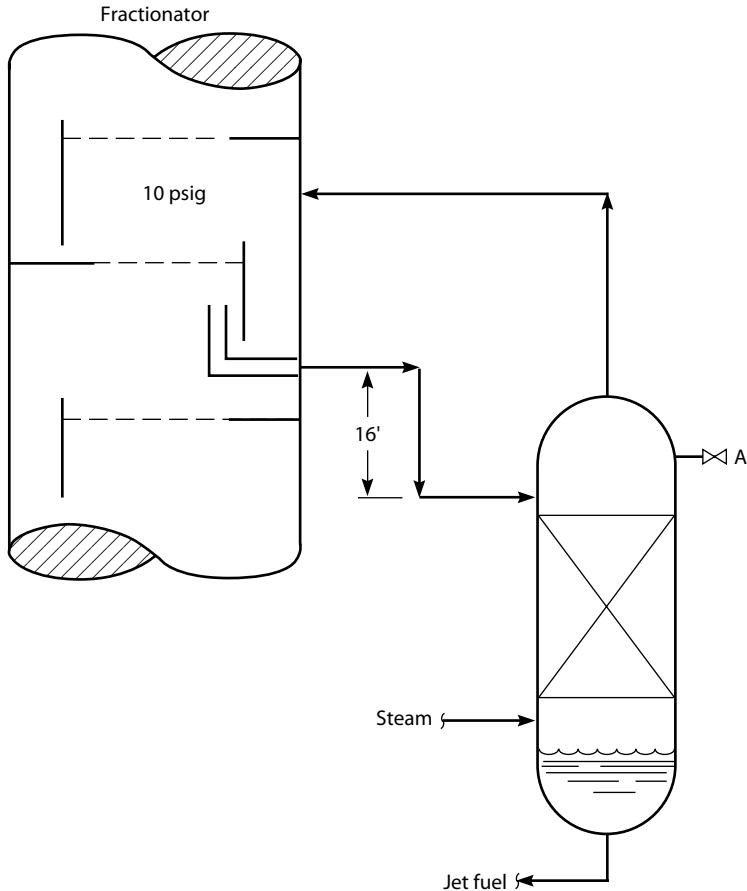


FIGURE 15.4 Hydraulics of a side-stream stripper.

The pressure head of the column of unstripped jet fuel 16 ft high, shown in Fig. 15.4, is

$$16 \text{ ft} \div 4 \text{ ft/psi} = 4 \text{ psi}$$

This means that there was a 4-psi pressure head driving force available to overcome the 3-psi pressure drop of the stripper's overhead vapor line. This was sufficient for the jet fuel to flow out of the fractionator and into the stripper.

Raising the steam flow to the stripper increased the pressure drop in the overhead vapor line from 3 to 5 psi. The pressure at point A in Fig. 15.4 then increased from 13 to 15 psig. The 4-psi pressure head driving force was not sufficient to overcome the 5-psi pressure drop of the stripper's overhead vapor line. The unstripped jet fuel could

no longer flow out of the fractionator and into the stripper, and the liquid level in the stripper was lost.

To correct this problem, I designed a larger-diameter vapor line connecting the stripper to the fractionator. Pressure drop through piping varies as follows:

$$\Delta P_{\text{new}} = \Delta P_{\text{old}} \left(\frac{\text{old pipe diameter}}{\text{new pipe diameter}} \right)^5$$

Changing the vapor line from a 3-in. pipe to a 4-inch pipe reduced the line's pressure drop from 3 to 0.7 psi. This permitted the stripping steam flow to be increased to the stripper, without impeding the jet fuel flow from the fractionator. The higher stripping steam flow efficiently removed the contaminant naphtha from the jet fuel product.

15.2.7 Liquid-Line ΔP

The liquid head driving force of 16 ft, or 4 psi, shown in Fig. 15.4, is actually not all available to overcome the higher stripper pressure. The frictional loss of the piping used to feed the stripper should be subtracted from the liquid head driving force. In the jet fuel example presented in Sec. 15.2.6, this frictional loss was neglected.

Sometimes the cheapest way to correct a hydraulic problem on a side-stream stripper is to pump the product into the stripper. I have modified several plants in this way with excellent results. But leaving the pressure in the stripper high will reduce stripping efficiency.

References

1. N. P. Lieberman, "Sour Water Strippers: Design and Operation," *Petroleum Technical Quarterly* (PTQ), Q2, 2013.
2. N. P. Lieberman, "Basic Technology Can Improve Refinery Profits," *Oil and Gas Journal*, July 18, 1994, pp. 50–54.

CHAPTER 16

Draw-Off Nozzle Hydraulics

Nozzle Cavitation Due to Lack of Hydrostatic Head

Imagine that many readers might skip this chapter. After all, a nozzle is simply a hole in a vessel, flanged up to a pipe (see Fig. 16.1). Why a whole chapter? Well, it is not that simple. Lots of process problems occur due to improperly designed draw-off nozzles.

16.1 Nozzle Exit Loss

The pressure drop of a fluid flowing through a nozzle is equal to

$$\Delta H = 0.34 \times V^2$$

where ΔH = pressure loss of the fluid as it flows through the nozzle,
in inches of fluid

V = velocity of the fluid, as it flows through the nozzle, in
feet per second

This equation assumes that before the fluid enters the nozzle, its velocity is small, compared to its velocity in the nozzle. The increase in the velocity, or the kinetic energy, of the fluid in the nozzle comes from the pressure of the fluid. This is Bernoulli's equation in action. The energy to accelerate the fluid in the draw-off nozzle comes from the potential energy of the fluid. This is Newton's second law of motion.

The coefficient used in the equation above (0.34) assumes the process fluid has a low viscosity. For most process nozzles, this is a

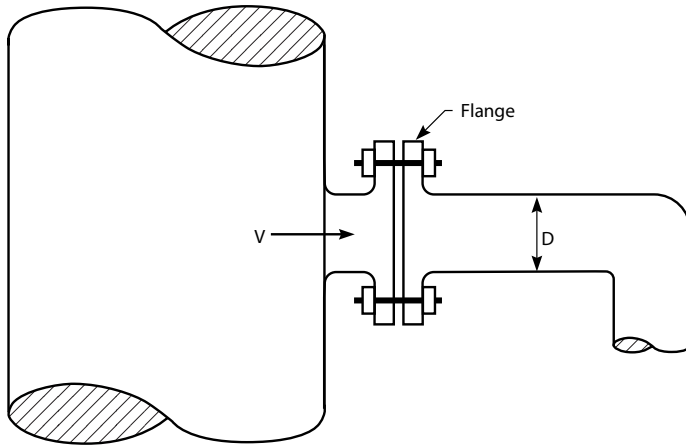


FIGURE 16.1 A draw-off nozzle.

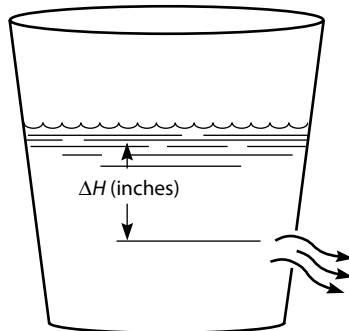
reasonable assumption. Detailed information on draw-off nozzle coefficients has been published in Crane.¹

For me, I don't trust published coefficients. I like to prove it myself: prove it by experimentation. Often, Liz (my coauthor) calls to me in the bathroom, "Norm, what are you doing? You've been in the bathroom forever."

What I'm doing is running water into the bathtub at a measured rate. I wait for the water to drain out until an equilibrium height is established. I have to get the height (DH) high enough to suppress vortexing. Then, using the diameter of the drain, I calculate the velocity (V) of the water flowing out the drain. I can then calculate the coefficient (0.34). It's a good thing I married Liz. The average woman might think I'm weird.

Let's take a close look at the use of the term ΔH . Figure 16.2 shows a bucket with a hole punched in its side. The velocity of the water as

FIGURE 16.2
 V is the velocity of the water escaping from the hole.



it escapes from the hole is V in the preceding equation. The height of water in the bucket above the center of the hole is ΔH . If we replaced the water in the bucket with gasoline, which has a lower density, the same hole velocity would occur with the same height of fluid.

The pressure drop of the fluid escaping from the bucket is called *nozzle exit loss*. Actually, nothing is lost; the potential energy, or pressure head of the water in the bucket, is just converted into velocity or kinetic energy.

16.1.1 Converting ΔH to Pressure Drop

A column of water 28 inch high exerts a head pressure of 1 psi, as shown in Fig. 16.3. To determine the pressure drop of water flowing through a hole, in pounds per square inch, we would calculate

$$\Delta P = \frac{0.34}{28 \text{ inch}} \times V^2$$

where ΔP = pressure drop in psi.

But perhaps we have gasoline flowing through the hole. The density of water is 62 lb/ft³; the density of gasoline is 40 lb/ft³. Since the weight, or pressure head, of a column of fluid is proportional to its density, we calculate

$$\Delta P = \frac{0.34}{28 \text{ inch}} \times \frac{40}{62} \times V^2$$

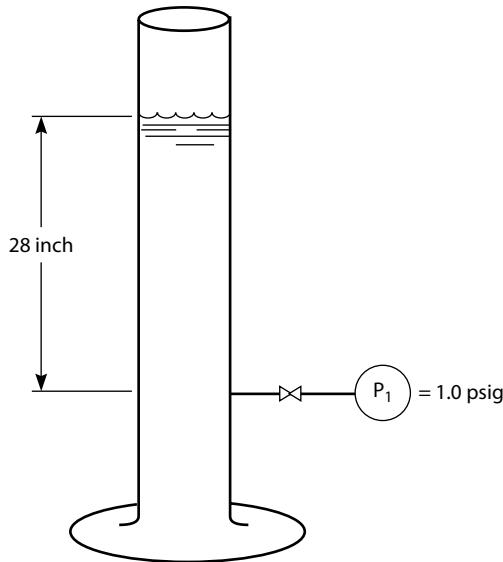


FIGURE 16.3 Water column exerting 1 psig of head pressure (1 psig = 28 inch of water).

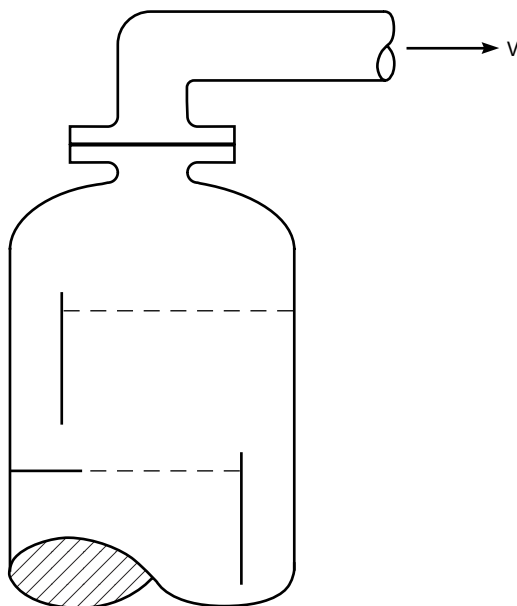


FIGURE 16.4 Nozzle exit loss for vapor flow.

Again, ΔP is the pressure drop, in psi, of the gasoline escaping through a hole in the bucket. If we have a vapor flowing out of the overhead nozzle on a fractionator, as shown in Fig. 16.4, the pressure drop of the vapor escaping from the vessel in the overhead vapor line is

$$\Delta P = \frac{0.34}{28 \text{ inch}} \times \frac{D_v}{62} \times V^2$$

where ΔP = pressure drop of a fluid, psi

D_v = density of a fluid such as a vapor, lb/ft³

V = velocity of fluid in nozzle, ft/s

This is an important equation to remember. It is the “pressure drop through a hole” equation. It works for water, steam, air, gasoline, alcohol, and all other fluids, unless viscosity is important. A viscosity of less than 5 or 10 centipoise (cP) or centistokes (cSt) (such as hot maple syrup) means that viscosity is not important.

16.2 Critical Flow

Everything I have just said is wrong when the fluid we are working with undergoes *critical flow*. First, critical flow applies only to compressible fluids such as vapor, steam, air, or gases. Second, as long as the pressure drop that we calculate through the nozzle is less than 20 or 30 percent of the upstream pressure, we can ignore critical flow.

Critical flow occurs when a compressible fluid velocity approaches the speed of sound: about 1000 ft/s. Process piping handling vapor is typically designed to work at a velocity of 100 ft/s.

16.3 Maintaining Nozzle Efficiency

16.3.1 Nozzle Limitations

Figure 16.5 is an exact representation of my bathtub. When my tub has a water level of 28 inch, the pressure right above the drain, at P_1 , is 1 psi. Having bathed, I now pull the plug. I also start the water running into the tub, to keep the water level at 28 in.

The velocity of water flowing from the tub through the drain is 20 ft/s. The pressure drop, in psi, of the water as it escapes from the tub, is

$$\Delta P = \frac{0.34}{28 \text{ inch}} \times (20)^2 = 5 \text{ psi}$$

The pressure at P_1 is now the 1-psi static head minus the 5-psi nozzle exit loss or negative 4 psig (or positive 10.7 psia). That is, the pressure at the drain is a substantial partial vacuum, or a *negative pressure*. By the term negative pressure I mean that it is below atmospheric pressure (atmospheric pressure at sea level is 16.7 psia).

This suggests that the pressure in a water drain can get so low that air could be sucked out of the bathroom and down the drain. Of course, we all see this happen several times a day—typically when we flush a toilet. So much air is drawn into the water drainage piping that we install vents on our roofs to release this air. The only requirement, then, for vapors to be drawn into a flowing nozzle is for the nozzle exit loss to be larger than the static head of liquid above the nozzle.

Incidentally, if a bird builds its nest on top of one of our roof toilet vents, we find the toilet will no longer flush properly. The experienced

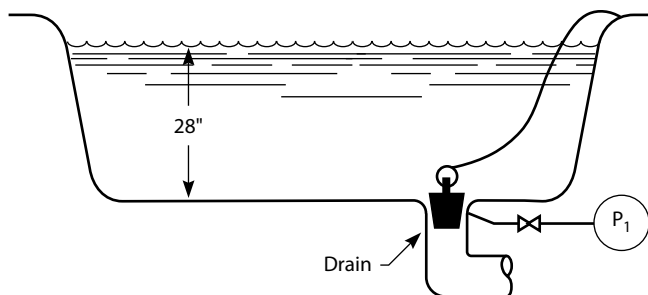


FIGURE 16.5 My bathtub.

plumber states that the toilet won't flush because it is suffering from "vapor lock," and this is true. A working knowledge of process equipment fundamentals often comes in quite handy around the home.

16.3.2 Effect of Bubble-Point Liquid

I have to confess to an odd habit. I sometimes bathe in boiling water. Draining a vessel full of liquid at its bubble or boiling point prevents the formation of a negative pressure, or partial vacuum, in the drain. As soon as the vacuum begins to form, the liquid will begin to generate vapor. This vapor would tend to restrict the flow of liquid as it exits the drain, and thus reduce its flow rate. As the velocity of the liquid flowing into the drain decreases, the pressure of the liquid at the inlet to the drain will increase. This increase in pressure will be just sufficient to suppress vaporization of the liquid in the drain (i.e., the liquid velocity will be reduced just enough to cause sufficient increase in pressure to prevent the liquid boiling).

Figure 16.6 illustrates the side draw-off of a fractionator. The liquid at point A is at its bubble or boiling point. We can be quite sure that this is true, because the liquid and vapor are in intimate contact at the vapor-liquid interface. We say that the vapor is at its dew point, and that the liquid is at its bubble point.

But how about the liquid at point B? Is this liquid also at its boiling or bubble point? It is the same liquid, having the same temperature and composition as the liquid at point A. But the pressure at point B is slightly higher than the pressure at point A.

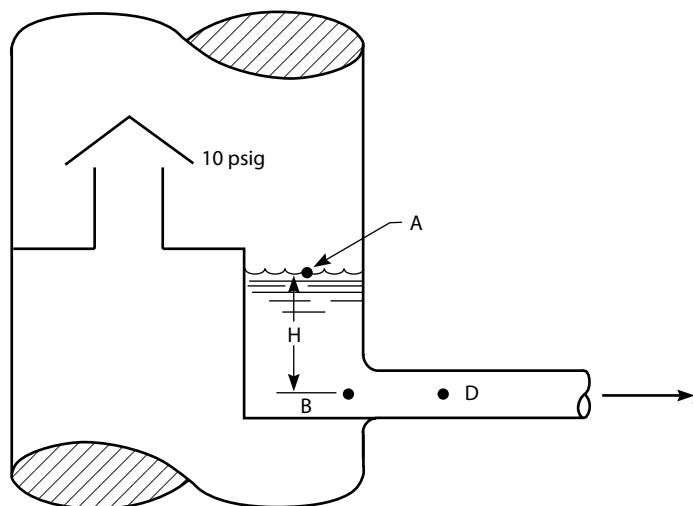


FIGURE 16.6 Fractionator side draw-off nozzle.

The pressure at point A is the pressure in the vessel: 10 psig. The pressure at point B is the pressure at point A plus the static head of liquid H, shown in Fig. 16.6. Let's assume that this liquid is water. Also, H equals 28 in. Then the pressure at point B is

$$10 \text{ psig} + \frac{28 \text{ inch}}{2.8 \text{ inch H}_2\text{O/psig}} = 11 \text{ psig}$$

The boiling point of water at 10 psig is about 240°F. This means that the temperature in the entire water draw-off sump is 240°F. But the pressure at point B is 1 psi above the water's bubble or boiling point. We say, then, that the water at point B is subcooled by an equivalent of 28 inch of liquid, or 1 psi.

But how about the pressure at point D? The liquid at both points B and D has the following in common:

- Same elevation
- Same fluid temperature
- Same fluid composition

If liquid is flowing through the nozzle, shown in Fig. 16.6, at, say, 9 ft/s, the pressure at point D will be lower than that at point B. Assuming the velocity in the draw-off sump to be close to zero, we can calculate the pressure at point D as follows: head loss due to increased velocity = $0.34 \times 9^2 = 28$ inch of water.

As 28 inch of water equals 1 psi of head pressure, the head loss due to increased velocity at the nozzle exit equals 1 psi. Thus, the pressure at point D will then be 11 psig minus 1 psi, that is, 10 psig.

16.3.3 Cavitation

For practice, calculate the pressure at point D assuming that the nozzle exit velocity is 10 ft/s. If you calculated 9.8 psig, you are likely wrong. You see, the water will start to flash to steam if it falls below its boiling-point pressure. The boiling-point pressure of water at 240°F is 10 psig. As steam is evolved at a lower pressure, the large volume of vapor chokes off the flow in the draw-off nozzle. The flow of water slows down. As the velocity of the water decreases, its pressure increases. The pressure will increase to 10 psig—that is, the boiling-point pressure of water at 240°F. At this pressure, the vaporization of the liquid in the draw-off nozzle is zero.

In summary, the lowest pressure that can be reached at point D in Fig. 16.6 is the pressure at point A. When these two pressures are equal, we say that the draw-off nozzle is limited by *cavitation*. If we were to lower the pressure downstream of point D, say, by opening a control valve, the increase in flow would be zero.

16.3.3.1 Vortex Breakers in Draw-Off Nozzles

The coefficient for nozzle exit loss of 0.34, shown above, assumes that there is no vortex breaker in the nozzle. Vortex breakers are certainly required if nozzle exit velocities exceed 4 ft/s. Vortex breakers are usually not needed if velocities in the draw-off nozzle are less than 2 ft/s.

As most such nozzles are designed at less than 2 ft/s, in general vortex breakers are not needed in most applications. However, many process plants have vortex breakers in all draw-off nozzles as part of their standard spec. This is bad for two reasons:

- Vortex breakers create turbulence, reduce the open area of nozzles, and thus lead to premature nozzle cavitation limits.
- Much more importantly, vortex breakers trap rags, gloves, and other trash left in towers during construction or turn-arounds. If a draw-off nozzle is equipped with both a down-comer seal pan and a draw-off box, then the draw-off nozzle is difficult, and oftentimes impossible, to inspect. Installing a vortex breaker that is not needed due to low draw-off velocities is just looking for trouble, without any justification.

If you note a trace of bitterness pertaining to the over zealous use of vortex breakers in my text, it is not without reason. The use of a totally unneeded vortex breaker in the bottom of a 90,000 BSD vacuum tower in the old ESSO LAGO Refinery in Aruba cost the new owners of the plant (Coastal) many millions of dollars in lost gas-oil production.

If you must use a vortex breaker, please consult my book, *Process Design for Reliable Operations*, to see how one can design a draw-off sump to prevent the vortex breaker from fouling.

16.3.4 External Restrictions

Figure 16.7 shows two almost identical draw-off arrangements. The only difference is the elevation of the control valve in the draw-off line. Control valve A is at the same elevation as the draw-off nozzle. The pressure drop across the control valve is 2 psi, or 56 inch of water. Let's assume that

- The increase in velocity through the draw-off nozzle is small, and hence the nozzle exit loss is zero.
- The frictional loss through the piping and nozzle is zero.

Even after we have made these two unlikely assumptions, the height of hot water in the draw-off sump must still be 56 inch above the center line of the draw-off nozzle. If not, the water would begin flashing to steam as it experienced a pressure drop of 2 psi flowing across the control valve. The evolved steam would then choke the

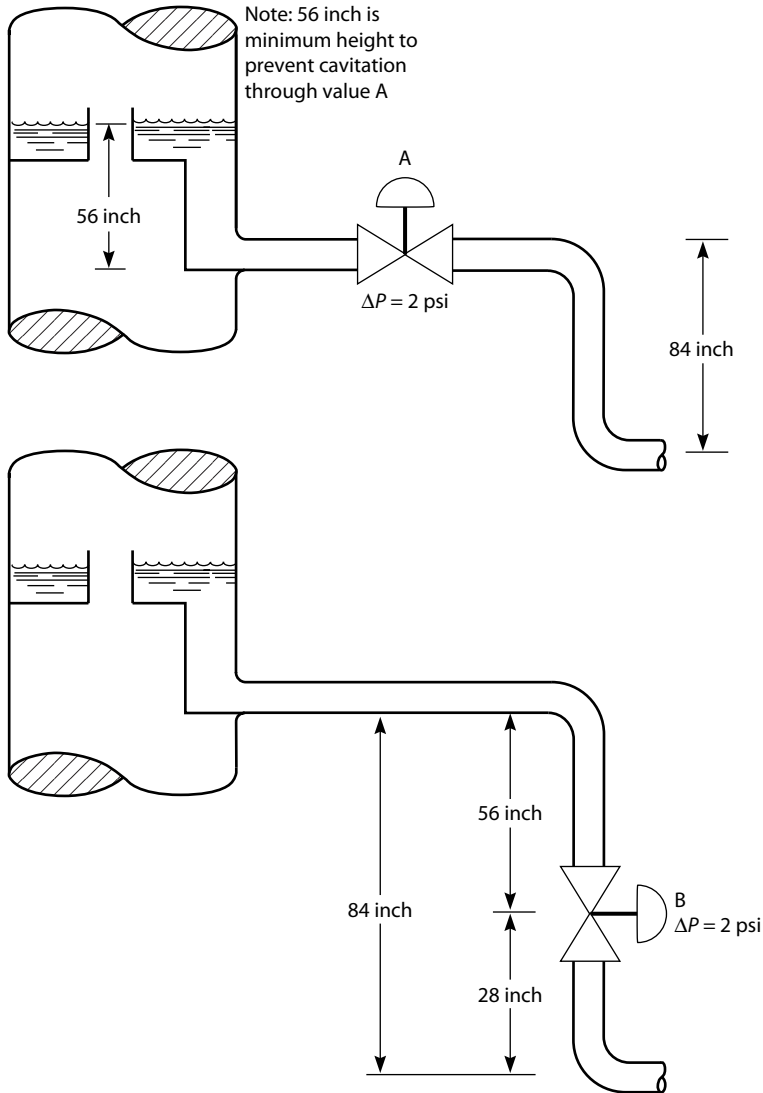


FIGURE 16.7 Location of control valves is critical.

water flow, reducing the pressure drop across the control valve until the pressure drop equaled the depth (or head) of water in the draw-off sump.

Control valve B in Fig. 16.7 is located 56 inch below the draw-off nozzle. The pressure drop across the valve is still 2 psi, or 56 in. of water. Again, we will neglect nozzle exit loss and friction loss. But in this case, the height of water in the draw-off sump may be zero. Why?

Referring to Fig. 16.7, the pressure head of water above control valve B is an extra 2 psi, or 56 inch of water. Thus, even if control valve B loses 2 psi of pressure, the water will not flash to steam. And this is true even when the sump is almost empty.

Does this mean that control valves should be located well below the elevation of the draw-off nozzle? Yes!

Does this mean that the amount of piping, fittings, gate valves, etc., should be minimized on a draw-off line, until the line drops 10 or 20 ft below the draw-off nozzle? Yes!

Does this mean any frictional losses due to external piping at the same elevation as the draw-off nozzle have to be added to the nozzle exit loss in determining the liquid level in the sump? Yes!

16.3.5 Increasing Flow from a Draw-Off Sump

If the flow is limited by nozzle exit loss, because the nozzle diameter is small or because the sump is shallow, no process change will increase the flow. However, if the restriction is outside of the vessel because of valves, fittings, or piping on the draw-off line, located at the same elevation as the draw-off nozzle, then there is an option to marginally increase flow.

Remove the insulation on the piping immediately downstream of the nozzle and spray cold water on the bare pipe. Reducing the temperature by a few degrees will subcool the liquid and suppress in-line vaporization or cavitation. This should permit an increase in the flow of product. It's the same idea as the common practice of operators spraying water on the suction of a hot pump to suppress pump cavitation as the flow rate is increased.

16.4 Overcoming Nozzle Exit Loss Limits

I once tried to increase the flow of jet fuel from a crude distillation column by opening the draw-off flow-control valve. Opening the valve from 30 to 100 percent did not increase the flow of jet fuel at all. This is a sure sign of nozzle exit loss—or cavitation limits. To prove my point, I increased the level of liquid in the draw-off sump from 2 to 4 ft. Since flow is proportional to velocity *and* head loss is proportional to (velocity)², the flow of jet fuel increased from 1000 to 1414 GPM. That is, the flow increased in proportion to the square root of the increase of liquid level in the sump. But the sump was only 4 ft high, and I needed to increase jet fuel flow even further. So I raised the pressure in the crude column from 10 to 14 psig. This increased the pressure in the draw-off sump by 4 psi.

The results of this increase in pressure are shown in Fig. 16.8. At first, the increase in pressure greatly increased the flow of jet fuel. But after a few minutes, jet fuel flow slipped back to its original rate. What happened?

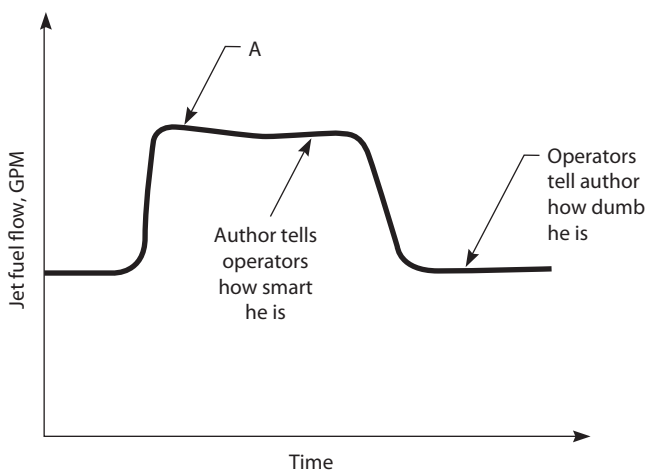


FIGURE 16.8 Equilibrium limits draw-off rates.

The problem is equilibrium. When the pressure in the tower was raised, lighter components from the vapor were forced to dissolve in the pool of liquid in the draw-off sump. This meant that, at the new equilibrium conditions, the lighter liquid would boil not at 10 psig but at 14 psig. But the composition of the liquid already in the sump did not change for a while. It took some time for the lighter liquid formed at the vapor liquid interface to displace the heavier liquid, already in the sump. During this period of time, jet fuel flow increased. But when the lighter liquid reached the draw-off nozzle, it began to vaporize. Flow was choked off by the evolving vapors, until the flow was reduced to the original rate. This is another symptom of a tower draw-off limited by nozzle exit loss or nozzle cavitation.

16.4.1 The Rule of Errors

This incident about the jet fuel draw-off happened at the Chevron refinery in 1987. It's true that I never solved the problem. It's also true that I rather made a fool of myself to the amusement of the operators. But still, I recall the incident as a happy memory.

The concept of the surface of a liquid pool reaching equilibrium with an altered vapor phase before the bulk of the liquid reaches equilibrium is a really useful idea. It's one of the reasons for a lag between a step change in an independent variable and the response of a dependent variable.

This incident got me thinking about the interrelationship between process dynamics and vapor-liquid equilibrium. So, even though I had made an error, I gained some important insights. Most of the things I've learned are through field observation.

It's like collecting snowflakes. I learned in school that no two snowflakes are identical. They're all six-pointed—but different. I had never seen a snowflake close up. So one snowy night I slipped out of my apartment in Brooklyn with a piece of black velvet. Sure enough, not only were all the snowflakes different, but each flake itself was a thing of wonder and joy.

Sitting in the office, you're warm and safe. Safe from making errors. But to progress, to learn, to advance the process and yourself, you've got to take chances. The chances are that if you go out to the plant and try a new idea, it won't work. Ninety percent of the time you will be wrong. But 100 percent of the time you'll learn something new.

16.4.2 Nozzle Exit Loss Calculation

My standard calculation for nozzle exit loss as shown at the start of this chapter is:

$$\Delta H = 0.34 (V^2) \quad (16.1)$$

where V = velocity, feet per second

ΔH = head converted to velocity, in inches, measured from the center of the nozzle

The coefficient of 0.34 is one that I've arrived at by measurement in my water barrel and bathtub, and then added a small safety factor for design purposes.

The actual theoretical calculation for converting head to acceleration is shown below:

- Step One—Let's assume that I have 1 lb of liquid flowing through a nozzle at a velocity of V feet per second.
- Step Two—To increase the kinetic energy of the liquid from no velocity to V requires:

$$(0.5)(M)(V)^2 \quad (16.2)$$

or since $M = 1$ pound of mass

$$(0.5)(V)^2 \quad (16.3)$$

- Step Three—I'll now divide by the gravitational constant at the surface of planet Earth (32.2 ft per second squared):

$$(0.5)(V) \text{ ft (sec)}^2 \div (32 \text{ ft/sec}^2) \quad (16.4)$$

or

$$(0.0155)(V)^2 \text{ ft} \quad (16.5)$$

- Step Four—To convert feet to inches of liquid, I'll multiply by 12:

$$(0.0156)(V^2) \text{ ft} \times (12 \text{ inch/ft}) \quad (16.6)$$

or

$$0.19(V^2) \text{ (inches)} \quad (16.7)$$

Then ΔH in inches of liquid equals:

$$\Delta H = 0.19(V^2) \quad (16.8)$$

The difference between the 0.19 coefficient in Eq. (16.8) and the 0.34 coefficient in Eq. (16.1) is due to friction losses through the nozzle and my safety factor used for design. All I have done with the above calculation is convert kinetic energy into potential energy. The purpose of this calculation is to demonstrate that there is a real theoretical basis for some of my basic rules of thumb. One of the best ways to improve one's engineering skills is to compare, on a quantitative basis, engineering theory and practice.

16.4.3 Pressure Recovery in Nozzles

If pressure is partially converted to velocity as a fluid accelerates through a nozzle, does this suggest that as a fluid enters a vessel through a nozzle, its velocity is partly converted back into pressure? Actually, I have observed this small effect in vacuum tower flash zone inlets, when I was measuring pressures to the nearest mm of Hg. Perhaps one-half to one-third of the energy used to accelerate a flow as a fluid enters a pipe from a vessel might be recovered when the fluid exits the pipe into a downstream vessel. The pressure not recovered is due to friction and turbulence. I almost always ignore nozzle pressure recovery in my calculations and field troubleshooting except in vacuum tower applications.

Reference

1. Crane, *Flow of Fluids through Valves, Fittings, and Pipe*, Technical Paper no. 410, 25th printing, 1991, Crane Company, Joliet, IL.

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CHAPTER 17

Pumparounds and Tower Heat Flows

Closing the Tower Enthalpy Balance

There are two ways to remove heat from a distillation tower: top reflux and circulating reflux. In this chapter, we call a circulating-reflux stream a *pumparound*.

The vast majority of fractionators have top reflux. Cold liquid from the reflux drum is pumped onto the top tray of the tower. The cold liquid flashes to a hotter vapor. For example, let's say 1500 lb/h of liquid butane at 100°F flashes to 1500 lb/h of vapor at 260°F.

The specific heat of butane is 0.6 Btu/[(lb)(°F)]. The latent heat of butane is 130 Btu/lb (*latent heat* means the heat needed to change a pound of liquid into a pound of vapor at the same temperature). The heat removed by the top reflux is

$$\begin{aligned}(260^{\circ}\text{F} - 100^{\circ}\text{F})(0.6)(1500) &= 144,000 \text{ Btu/h} = \text{sensible heat} \\ &+ \\ (130)(1500) &= 195,000 \text{ Btu/h} = \text{latent heat} \\ \hline 339,000 \text{ Btu/h} &= \text{total heat removed}\end{aligned}$$

17.1 The Pumparound

17.1.1 Pumparound Heat Removal

Figure 17.1 shows an alternate method, called *circulating reflux* or *pumparound*, to remove heat from a tower. Hot liquid, at 500°F, is drawn from tray 10, which is called the *pumparound draw tray*. The liquid pumparound is cooled to 400°F. The cooled liquid is returned to the tower at a higher elevation onto tray 9. It appears from Fig. 17.1 that the cold 400°F pumparound return liquid is entering the downcomer from tray 8. This is often good design practice. Tray 9 is called the *pumparound return tray*.

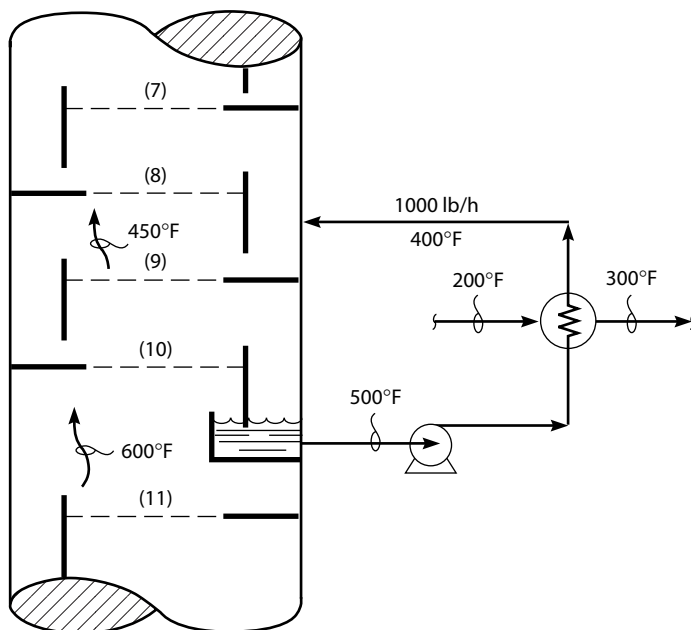


FIGURE 17.1 A pumparound or circulating reflux.

The purpose of the pumparound is to cool and partially condense the up-flowing vapors. The vapors to pumparound tray 10 are at 600°F. The vapors from the pumparound return tray 9 are at 450°F. There are two pumparound trays (9 and 10) in the column. This is the minimum number used. A typical number of pumparound trays is 2 to 5.

Let's calculate the heat removed in the pumparound circuit shown in Fig. 17.1. Assume that the specific heat of the pumparound liquid is 0.7 Btu/[(lb)(°F)]:

$$(500^{\circ}\text{F} - 400^{\circ}\text{F}) \times (0.7) \times 1000 \text{ lb/h} = 70,000 \text{ Btu/h}$$

I have not shown the flow of liquid on the cold side of the pumparound heat exchanger in Fig. 17.1, but we can calculate its flow. Let's assume that the specific heat of the cold-side liquid is 0.5 Btu/[(lb)(°F)]. Then the cold-side (or shell-side) flow is

$$\frac{(70,000 \text{ Btu/h})}{(300^{\circ}\text{F} - 200^{\circ}\text{F}) \times [0.5 \text{ Btu}/(\text{lb})(^{\circ}\text{F})]} = 1400 \text{ lb/h}$$

What I have just illustrated is the most powerful tool in my bag of tricks—calculating a flow from a heat balance.

17.1.2 Purpose of a Pumparound

Why do we wish to remove heat from the vapor flowing up through tray 10? The circulating pumparound is cooling the vapor flowing

through tray 10 from 600 to 450°F as it leaves tray 9. But for what purpose? Figure 17.2 shows the rest of the tower we have been discussing. Note that the tower-top reflux flow is controlling the tower-top temperature. If we were to reduce the pumparound circulation rate, less heat would be extracted from trays 9 and 10. More and hotter vapor would flow up the tower. The top reflux temperature control valve would open. The top reflux rate would go up. The vaporization of reflux on the top tray would increase. The overhead condenser duty would increase.

The decrease in the heat duty of the pumparound heat exchanger would equal the increase in the heat duty of the overhead condenser. Thus, we say that the *heat balance* of the tower is preserved. Some of the heat that was being recovered to the cold fluid, shown in Fig. 17.2, is now lost to cooling water in the overhead condenser. This shows the most important function of pumparounds: recovering heat to a process stream that would otherwise be lost to the cooling tower.

Let's say that the cooling-water outlet temperature from the condenser was 140°F. This is bad. The calcium carbonates in the cooling water will begin to deposit as water-hardness deposits inside the tubes.

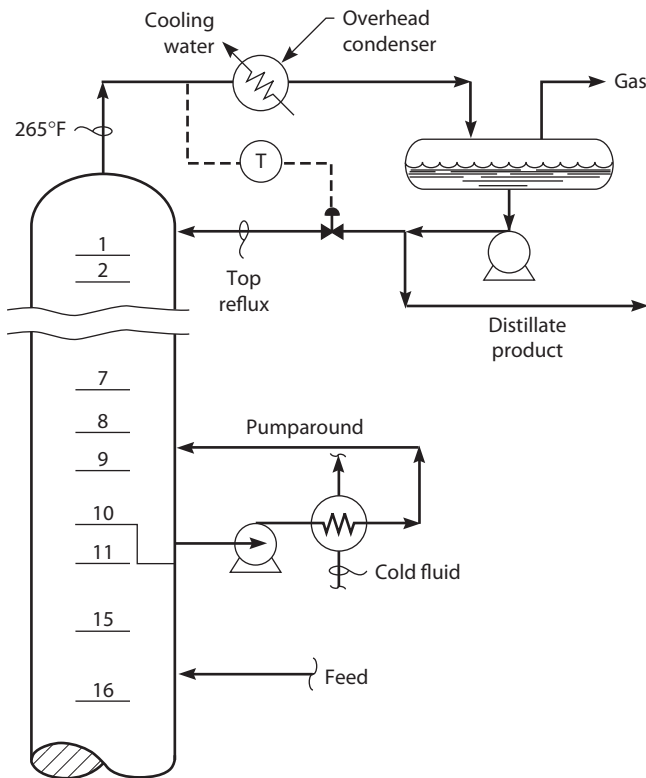


FIGURE 17.2 Two ways of removing heat from a tower.

It is best to keep the cooling-water outlet temperature below 125°F to retard such deposits. Increasing the pumparound heat removal will lower the cooling-water outlet temperature.

Another purpose of the pumparound is to suppress top-tray flooding. If tray 1 or 2 in Fig. 17.2 floods, the operator would observe the following:

- The tower-top temperature would increase.
- The distillate product would become increasingly contaminated with heavier components. If this were a refinery crude fractionator, we would say that the endpoint of the naphtha overhead product would increase.
- The pressure drop across the top few trays would increase.
- The liquid level in the reflux drum would increase.

If the operator increases the reflux rate to reduce the tower-top temperature, the top temperature will go up rather than down. This is a positive indication of *top-tray flooding*. The correct way to suppress top-tray flooding is to increase the pumparound duty. This can be done by increasing the cold-fluid flow through the pumparound heat exchanger, or the pumparound flow itself could be increased. Either way, the flow of vapor flowing up to tray 8 will decrease. The flow of vapor through trays 1 to 7 will also decrease. The low vapor velocity will reduce the tray pressure drop. The ability of the vapor to entrain liquid will be reduced. The height of liquid in the downcomer will be reduced, and tray flooding will be suppressed.

Increasing pumparound heat duty will unload the overhead condenser. This will cool off the reflux drum. A colder reflux drum will absorb more gas into the distillate product. Less gas will be vented from the reflux drum, and this is often desirable.

Finally, it is best remembered that, as we said earlier in this chapter, heat recovered in the pumparound heat exchanger is often a valuable way to recover process heat. Heat not recovered in the pumparound exchanger is lost to cooling water in the overhead condenser.

17.1.3 Do Pumparounds Fractionate?

The process design engineer typically assumes that a pumparound is simply a way to extract heat from a tower. The engineer does not expect the trays used to exchange heat between the hot vapor and the cold liquid to also aid in fractionation. In practice, this is not what happens.

Let's refer back to Fig. 17.1. Note that the vapor temperature leaving tray 9 is 450°F. The temperature of the liquid leaving tray 10 is 500°F. This sort of temperature difference shows that fractionation is

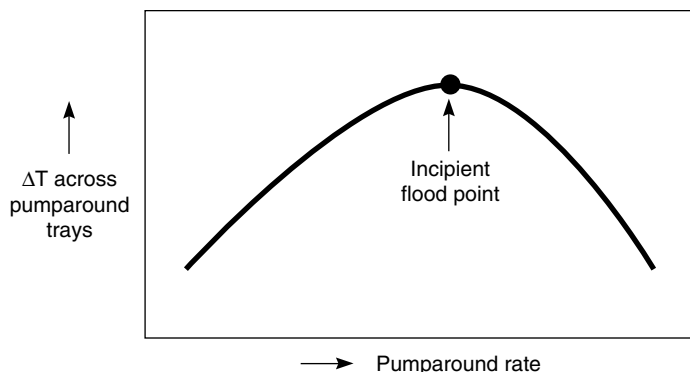


FIGURE 17.3 Pumparound trays do fractionate.

taking place across the pumparound trays. The temperature difference between

$$\Delta T = (\text{temperature of liquid leaving a lower tray}) \\ \text{minus } (\text{temperature of vapor leaving a higher tray})$$

is a measure of the amount of fractionation. The bigger this temperature difference, the more the fractionation that is taking place across the trays. I have run plant tests that indicate that increasing pumparound circulation rates increases this temperature difference—up to a point. However, if the circulation rate is increased past this point, then the temperature difference is reduced. This idea is expressed best in Fig. 17.3. As the pumparound rate is increased, tray efficiency is improved. However, at some point, the pumparound liquid flow becomes too great. Probably, at this point the downcomers start to back up. Tray efficiency is impaired because of this downcomer flooding. The temperature difference between the liquid leaving the pumparound draw tray minus the temperature of the vapor leaving the pumparound return tray becomes smaller. This point is called the *incipient flood point* for the pumparound trays.

17.2 Vapor Flow

17.2.1 How Top Reflux Affects Vapor Flow

Let's assume that the vapor flow into a tower is constant. Figure 17.4 shows such a situation. Both the pounds per hour and temperature of the vapor flowing up to tray 9 are constant. This means that the heat flow into the tower is constant.

Now, let's increase the reflux rate. Certainly, the result will be

- The tower-top temperature will decrease.
- The gasoline overhead product flow will also decrease.

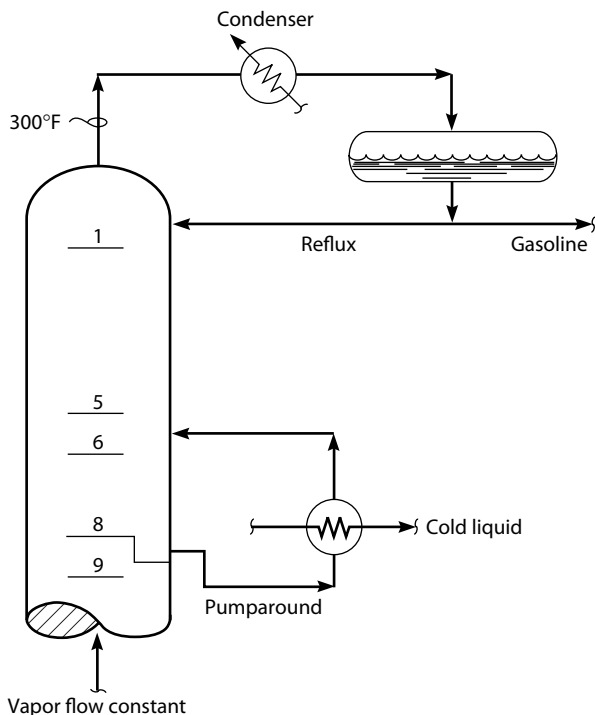


FIGURE 17.4 Effect of top reflux on vapor flow.

But what will happen to the flow of the vapor leaving tray 3, 4, or 5? I ask this question assuming that the pumparound heat duty is fixed. Will the pounds per hour of vapor flowing through trays 3, 4, or 5

- Increase?
- Decrease?
- Remain the same?

The correct answer is that the weight flow of vapor will increase! Surprised? Most people are. Let me explain.

When we increase the reflux rate, the tower-top temperature drops—let's say from 300 to 240°F. Actually, the temperature of the vapor leaving all the trays in the tower will decrease. The effect is bigger on the top tray and gradually gets smaller as the extra reflux flows down the tower. If the top-tray temperature has dropped by 60°F, then the vapor temperature leaving tray 9 might drop by only 5°F. Let's assume that the extra reflux causes the temperature of the vapor from tray 4 to decrease by 40°F. We can say that the sensible-heat content of the vapor has decreased. *Sensible heat* is a measure of the heat content of a vapor due to its temperature. If the specific heat of the vapor is 0.5 Btu/[lb)(°F)], then

the decrease in the sensible-heat content of the vapor when it cools by 40°F is 20 Btu/lb.

A small portion of this 20 Btu is picked up by the increased liquid flow leaving tray 4. The main portion of this heat is converted to *latent heat*. This means that some increment more, of the liquid on the tray turns into a vapor. But where does this extra liquid, which vaporizes on the tray, come from? It comes from the extra top reflux.

The vaporization of the extra reflux cools the tray. The extra vapor generated adds to the vapor flow from the tray. This increases the vapor flow from the tray. Even though the heat flow into the tower is constant, increasing the top reflux does increase the pounds of vapor flowing up the tower.

17.2.2 Reflux Effect on Vapor Molecular Weight

Let's assume the following for the tower shown in Fig. 17.4:

- Constant vapor flow to tray 9
- Tower-top reflux flow increased
- Pumparound duty is constant

As a result of the increased reflux rate, the

- Tower-top temperature drops
- Gasoline flow drops

This is the same problem I discussed previously. It will result in an increase in the mass flow of the vapor through trays 3, 4, and 5. But what will happen to the *molecular weight* of the vapor? Will it increase, decrease, or remain the same?

The answer is that the vapor's molecular weight will decrease as the reflux rate is increased. But why?

The vapor leaving each tray is in equilibrium with the liquid. This means that the vapor leaving each tray is at its dew point and the liquid leaving each tray is at its bubble point. As the top reflux rate is increased, all the trays are cooled. The vapors leaving trays 3, 4, and 5 are cooled. As a vapor at its dew point cools, the heavier components in the vapor condense into a liquid. The remaining vapors have a lower molecular weight because they are lighter. But this is only half the story. Let us continue!

As the heavier components in the vapor condense into a liquid, they give off heat. This heat is called the *latent heat of condensation*. This latent heat is picked up by the liquid flowing across the tray. This liquid flow is called the *internal reflux*. This latent heat promotes extra vaporization of the internal reflux. Naturally, the lighter, lower-boiling-point components preferentially vaporize from the internal reflux. These lighter components have a relatively low molecular weight.

The uncondensed vapors flowing from the tray below, plus the newly generated vapors from the reflux, flow to the tray above. The combined molecular weight of vapors is thus reduced. As the molecular weight decreases, the volume of each pound of vapor increases.

As the molecular weight of the vapor decreases, the density of the vapor decreases. As the density of a vapor is reduced, each pound of vapor occupies more volume. It is true that cooler vapors do occupy less volume per pound than do warmer vapors of the same composition. That is, gases and vapors tend to contract on cooling and expand on heating. However, this is a small effect compared to the increase in the volume of vapors, due to the decrease in the vapor's molecular weight.

17.2.3 Reflux Effect on Vapor Volume

Now, let's summarize the effect on vapor flow due to increased reflux. Again, we are considering trays 3, 4, and 5 in Fig. 17.4:

1. Increasing the top reflux increases the pounds of vapor flow.
2. Increasing the top reflux increases the volume per pound of the vapor.
3. Increasing the tower-top reflux rate increases the rate (in ft^3/s) of vapor flow through the trays, because of the combined, additive effect of factors 1 and 2.

How, then, does increasing the top-tray reflux rate affect the vapor and liquid loading on the trays below? Obviously, the liquid flow rate increases. But so does the vapor flow rate.

Does this mean that trays 3, 4, and 5 could flood, because the top reflux rate was increased? Yes!

Does this mean that we could flood these trays without increasing the heat flow into the tower? Yes!

Does this mean that we could flood these trays without increasing the pounds of vapor flow into the tower, simply by raising the reflux rate? Absolutely correct!

If an increase in the tower-top reflux rate causes the top of the tower to flood, how should the operator respond? She should then increase the pumparound flow to reduce the pounds of vapor flow to tray 5, as shown in Fig. 17.4. But suppose this causes the pumparound trays 6, 7, and 8 to flood, because of the extra liquid flow? She should increase the cold liquid flow through the pumparound heat exchanger. If this cannot be done either, the tower pressure can be increased. This will increase the density of the flowing vapors and shrink the volume of the vapors that the trays must handle.

The effect of increasing the reflux rate on a crude tower, even with a constant condenser duty, will be to increase the condenser pressure drop. This happens because the volume and the weight of the tower overhead vapors will increase as the tower-top temperature drops. At the Calumet Refinery in Shreveport, Louisiana, the crude tower reflux rate was increased by 25 percent and the tower-top temperature dropped by 50°F–60°F. The overhead condenser ΔP increased from about 12 to 24 psi. The tower pressure was kept constant, but the reflux drum pressure declined from 17 to 5 psig.

The mechanism that accounts for this is twofold:

1. Reducing the tower-top temperature converts some of the heat content of the vapor to latent heat of vaporization of the top reflux.
2. The molecular weight of the vapors leaving the top of the tower is reduced.

17.3 Fractionation

17.3.1 Improving Fractionation

Here is our problem. A refinery crude distillation tower is producing gasoline, truck diesel, and a gas oil. The diesel is contaminated with the gas oil. Also, the gas oil is contaminated with the lighter diesel. As shown in Fig. 17.5, the vapor flow into the tower is constant. Our job is to improve the *degree of fractionation* between the diesel and gas oil. Our objective is to remove the relatively heavy gas oil from the diesel and to remove the lighter diesel from the gas oil.

We could reduce the amount of diesel product drawn from the tower. That could wash the heavier gas oil out of the diesel. But it would also increase the amount of diesel in the gas oil. Increasing the heat removed in the pumparound would have a similar effect: less gas oil in diesel, but more diesel in gas oil.

How about decreasing the heat removed in the pumparound? This would seem to allow the lighter diesel to more easily vaporize out of the gas-oil product. But will this action result in increasing the contamination of diesel with heavier gas oil? This answer is no—but why not?

Reducing the pumparound heat-removal duty increases the vapor flow from tray 8 in the column shown in Fig. 17.5. The extra pounds of vapor flow up the tower and raise the tower-top temperature. The reflux control valve opens to cool the tower-top temperature back to its temperature set point. Then the liquid flow rates from trays 1, 2, and 3 onto tray 4 all increase. If the diesel draw-off rate is maintained constant, the liquid overflow rate onto trays 5, 6, and 7 will increase. This liquid flow is called the *internal reflux*. Trays 5, 6, and 7 are the

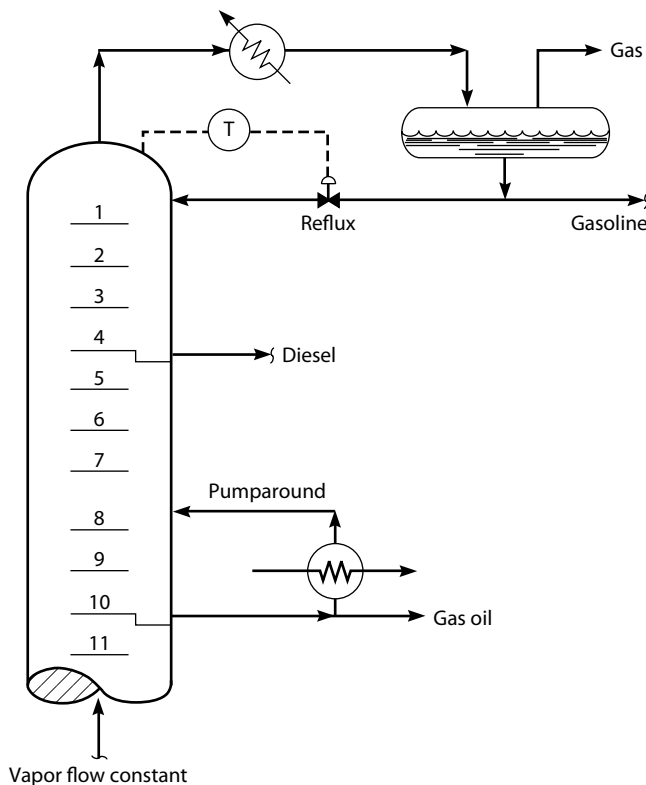


FIGURE 17.5 Decreasing pumparound improves fractionation.

trays that fractionate between diesel and gas oil. The more efficiently they work, the less the contamination of the adjacent products.

The way we increase the fractionation efficiency of trays is to make the trays work harder. The correct engineering way to say this is: "To improve the separation efficiency between a light and heavy product, the vapor flow rate through the trays is increased, and the internal reflux flowing across the trays is increased."

Again, this improvement in the degree of fractionation developed by trays 5, 6, and 7 is a result of reducing the amount of heat duty removed by the pumparound flowing across trays 8, 9, and 10.

17.3.2 Flooding the Fractionation Trays

Reducing the pumparound duty increases the tray loadings on trays 1 through 7. But in so doing, the trays operate closer to their incipient flood point. This is fine. The incipient flood point corresponds to the optimum tray performance. But if we cross over the incipient flood point, and trays 5, 6, and 7 actually start to flood, their fractionation

efficiency will be adversely affected. Then, as we decrease the pump-around heat-removal duty, the mutual contamination of diesel and gas oil will increase.

From an operating standpoint, we can see when this flooding starts. As we decrease the pumparound duty, the temperature difference between the diesel- and gas-oil product draws should increase. When these two temperatures start to come together, we may assume that we have exceeded the incipient flood point, and that trays 5, 6, and 7 are beginning to flood.

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CHAPTER 18

Condensers and Tower Pressure Control

Hot-Vapor Bypass: Flooded Condenser Control

The total condensation of a vapor to a liquid is best illustrated by the condensation of steam to water. Figure 18.1 is a rather accurate reproduction of the radiator that heated my apartment in Brooklyn. Steam flowed from the boiler in the basement. The steam condensed inside the radiator and flowed back into the boiler through the condensate drain line. This is a form of thermosyphon circulation. The driving force for the circulation is the differential density between the water in the condensate drain line and the steam supply line to the radiator (see Chap. 8, for a discussion of thermosyphon reboilers).

The bigger the radiator, the more heat is provided to a room. The bigger the radiator, the faster the steam condenses to water inside the radiator. A larger radiator has more *heat-transfer surface area* exposed to the condensing steam. Unfortunately, the radiator shown in Fig. 18.1 is suffering from a common malfunction. Water-hardness deposits have partly plugged the condensate drain line. Calcium carbonate is a typical water-hardness deposit.

Anyone who has lived in an apartment house in Brooklyn will have had to occasionally bang on the radiator to increase heat flow. Banging on the radiator often breaks loose the carbonate deposits in the condensate drain line. The steam condensate, which has backed up in the radiator, now empties. This exposes more of the interior surface area of the radiator to the condensing steam.

It rather seems that 40 percent of the surface area of the radiator in Fig. 18.1 is submerged under water. If the water is drained out, does this mean that the rate of steam condensation will increase by

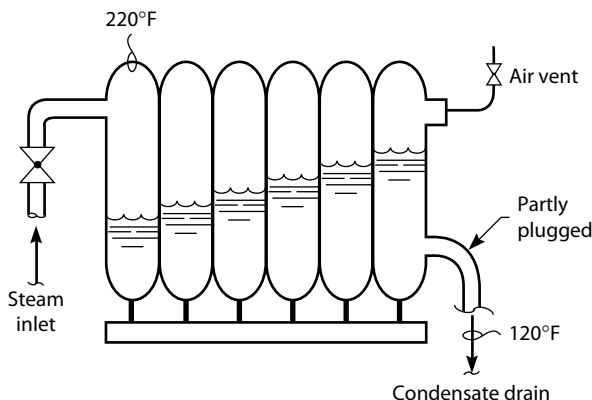


FIGURE 18.1 Radiator in a Brooklyn apartment house, circa 1946.

the same 40 percent? Answer—yes! Does this mean that the radiator heat-transfer duty will increase by 40 percent? Answer—not quite.

18.1 Subcooling, Vapor Binding, and Condensation

18.1.1 Subcooling

18.1.1.1 Effect of Subcooling

When steam condenses at atmospheric pressure, it gives off 1000 Btu/lb of condensing steam. This is called the *latent heat of condensation of steam*.

When water cools off from 220 to 120°F, it gives off 100 Btu/lb of water. This heat represents the sensible-heat content of water between 220 and 120°F.

It takes less of the radiator's surface area to condense 1 lb of steam at 220°F than to cool off 1 lb of water from 220 to 120°F. And this is true even though the condensation of steam generates 10 times as much heat as the cooling of hot water.

Does this mean that it is a lot easier to condense steam than to cool water? Yes. This also explains, then, why condensate backup reduces the rate of heat transfer and condensation.

18.1.1.2 Mechanics of Subcooling

As the condensed steam flow out of the radiator is restricted, the surface area of the radiator available to cool the hot water increases. Hence, the water temperature leaving the radiator decreases. To summarize, the effect of restricting the condensate flow from a radiator or condenser is to

- Build the water level in the radiator
- Reduce the rate of latent-heat transfer from the steam

- Increase the rate of sensible-heat transfer from the condensate
- Reduce the overall heat-transfer duty from the radiator

Incidentally, the correct way to remove hardness deposit is by chemical cleaning. Violent banging on radiators is considered bad form in South Brooklyn.

18.1.2 Air Lock

Vapor binding, or air lock, is another common cause of household radiator malfunction. Often, the vapor accumulating in the radiator is CO_2 , rather than air. The CO_2 originates from the thermal decomposition of carbonates in the boiler. Regardless, air and CO_2 form a non-condensable vapor in the radiator. These non-condensables mix with the steam in the radiator. The non-condensables then reduce the concentration of the steam by dilution. The diluted steam has a lower partial pressure than pure steam. The lower the partial pressure of the steam, the more difficult it is to condense. As the rate of condensation of the steam drops, so does the heat radiated by the radiator.

In refinery steam condensation services (in both reboilers and steam turbines exhausting to vacuum surface condensers), I have observed that small amounts of non-condensables greatly reduce the apparent heat-transfer coefficient.

In reboilers, propane or butane may leak into the lower-pressure steam side of the reboiler. In surface condensers, air could be drawn into the condenser's shell, which operates under vacuum conditions. In the latter case, I've observed that the heat-transfer coefficient, which should be $200 \text{ Btu/hr/ft}^2/^{\circ}\text{F}$, is only about $50 \text{ Btu/hr/ft}^2/^{\circ}\text{F}$.

It seems these non-condensables are increasing the heat-transfer film resistance by interfering with the ability of the steam molecules to come into contact with the cooler heat-transfer surface area. A tiny amount of air in a steam condenser will reduce the rate of the steam condensation by an order of magnitude (see Chaps. 24 and 25).

To restore the efficiency of a radiator suffering from the accumulation of non-condensables inside its condensing coils, the non-condensable gases have to be removed. The air vent shown in Fig. 18.1 serves this purpose.

18.1.2.1 Explanation of Vapor Binding

We can measure the non-condensable concentration in the flow of a mostly condensable vapor such as steam. However, the actual concentration of non-condensables in contact with the condensate layer around the tube wall or inside the tube is much greater. While the condensable vapor condenses and drains away as a liquid phase, the only way the non-condensables can escape from the tube surface is by diffusion backwards against the bulk of the vapor flow. In order to make this diffusion possible, there must be a concentration gradient

with a higher non-condensable concentration near the tube surface, which then impedes the condensation rate of the bulk of the vapor flow (see R. S. Silver, *Proc. Inst. Mech. Engineers*, Vol. 178, p. 351, 1963–1964). This explains why a small amount of venting non-condensables often has the dramatic effect of increasing condensation rates, especially when condensing pure components such as steam.

I also believe that this explains a problem that I've seen several times in steam turbine exhaust vacuum surface condensers. That is, an apparent large loss in the steam heat-transfer coefficient, due to a relatively small air leak into the vacuum surface condenser's shell.

To summarize, the two most common malfunctions of a steam condenser (or radiator) are

- Condensate backup
- Non-condensable accumulation

And these two malfunctions are also the most common problems we encounter in the design and operation of shell-and-tube heat exchangers used in total condensation service for hydrocarbons, as well as for steam.

18.1.3 Condensation and Condenser Design

18.1.3.1 Condensation in Shell-and-Tube Heat Exchangers

If what you have just read seems to be a repetition of the discussion of steam reboilers in Chap. 12, your analysis is correct. A steam reboiler has the same problems and works on the same principles as a process condenser. The only difference is that a steam reboiler's heat is removed by the shell-side process fluid, and a process condenser's heat is removed by cooling water or air.

Figure 18.2 is a sketch of a depropanizer overhead condenser. Let's make a few assumptions about this shell-and-tube condenser:

- The propane is totally condensed as it enters the reflux drum.
- There is no vapor vented from the reflux drum, but there is a vapor-liquid interface in the drum.
- The reflux drum is elevated by 20 ft above the top of the condenser.
- We are dealing with pure (100 percent) propane.

If the liquid level in the reflux drum is located 20 ft above the condenser, does this mean that the liquid level in the pipe feeding the condenser is also 20 ft above the condenser? Is this possible? No!

If the pipe to the condenser maintained a liquid level, then the shell side of the condenser would be full of propane. But if the shell side of the condenser were really liquid full, the tubes would not

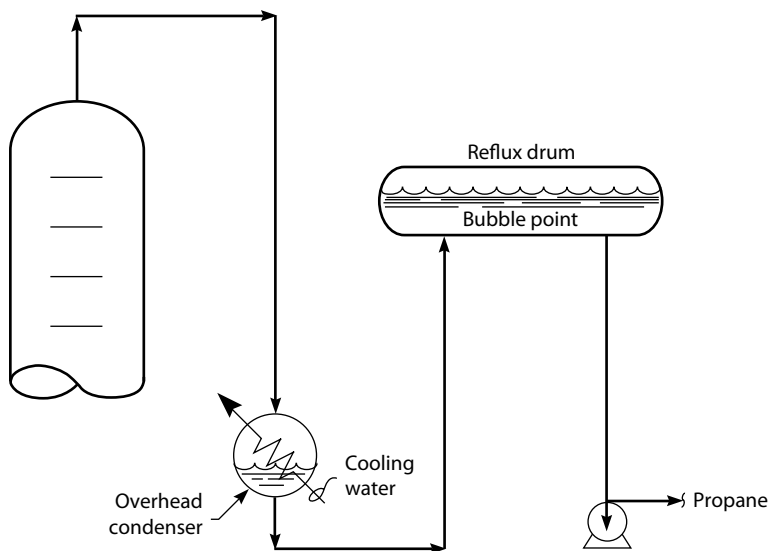


FIGURE 18.2 Total condensation below the reflux drum.

contact the vapor. If the tubes do not contact the vapor, then the rate of condensation is zero. Perhaps a small amount of heat transfer would take place as the liquid propane became subcooled. But none of the propane vapor would condense.

Therefore, the liquid level in the overhead condenser would have to be somewhere in the condenser's shell. But then the liquid in the condenser would be below the reflux drum. How, then, does the liquid get from the lower elevation of the condenser to the higher elevation in the reflux drum? We will have to explain this hydraulic problem later. But for now, we can say that most reflux drums are elevated 20 or 30 ft above grade to provide *net positive suction head* (NPSH) for the reflux pump. Also, most shell-and-tube condensers are located at grade for easier maintenance during unit turnarounds.

18.1.3.2 Subcooling in a Shell-and-Tube Condenser

Figure 18.3 is the same propane condenser shown in Fig. 18.2. Let's assume that the pressure drop through the shell side is zero. Again, we are dealing with a pure component: propane. The inlet vapor is at its dew point. That means it is *saturated vapor*. Under these circumstances, the outlet liquid should be a *saturated liquid*, or liquid at its bubble point. As the inlet dew-point temperature is 120°F, the outlet bubble-point temperature should be 120°F. But, as can be seen in Fig. 18.3, the outlet shell-side liquid temperature is 90°F, not 120°F. Why?

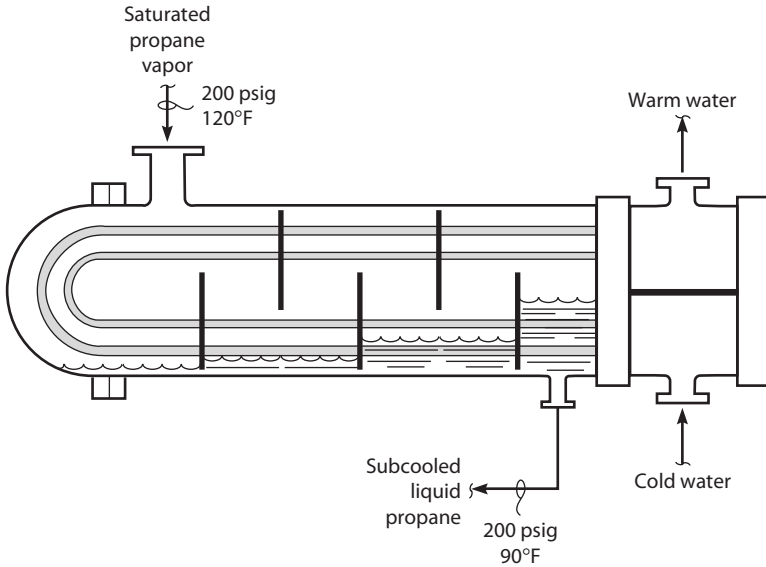


FIGURE 18.3 Condensate backup in a shell-tube heat exchanger.

The reason is condensate backup. The condensate backup causes subcooling; that is, the liquid is cooled below its bubble point or saturated liquid temperature. Perhaps a rat has lodged in the condensate outlet pipe. The rat restricts condensate drainage from the shell side. To force its way past the dead rat, the propane backs up in the condenser. The cold tubes in the bottom of the shell are submerged in liquid propane. The liquid propane is cooled below its bubble-point temperature.

Note that the propane vapor is still condensing to propane liquid at 120°F. The condensed liquid is in intimate contact with the propane vapor as it drips off the outside surface of the colder condenser tubes. The saturated propane vapor condenses directly to saturated propane liquid at 120°F. The saturated, or bubble-point, liquid then drips from the condensation zone of the condenser into the subcooling zone of the condenser. This is the zone where the tubes are submerged in liquid.

Try running your hand along the outside of such a condenser. Feel for the point on the surface of the shell where there is a noticeable drop in temperature. The upper part of the shell will be hot. The lower part of the shell will be cold. The transition point corresponds to the liquid level of condensate in the shell. The condensate level will always be higher toward the shell outlet nozzle. Again, this all applies only to condensers in total condensing service.

18.1.3.3 Effect of Condensate Backup

When the condensate level in an exchanger increases, the area of the condenser devoted to subcooling the condensate increases. But the area of the exchanger available for condensing decreases. That is bad!

When the area of the exchanger available for condensing is reduced, the ease of condensation is also decreased. Depending on the circumstances, one of two unfavorable things will now happen:

1. If the supply pressure of the condensing vapor is fixed, the rate of condensation of the vapor will fall. In an apartment house in Brooklyn, this means that your bedroom will get cold.
2. If the condensing vapor flow rate is fixed, the condensation pressure will increase. If you are operating a debutanizer in Texas City, this means that the tower's safety release valve will pop open and release a cloud of butane over Lamar University.

Heat removed by condensation is easy. The heat-transfer coefficient U for condensation of pure, clean vapors may be 400 to 1000 Btu/hr/ft² of heat-exchanger surface area, per °F of temperature-driving force. The U value for subcooling stagnant liquid may be only 10 to 30. Condensate backup is the major cause of lost heat transfer for heat exchangers in condensing service.

18.1.3.4 Reflux Drum Elevation Increase Promotes Subcooling

Once upon a time many years ago, a tragic event occurred in Louisiana. A rat entered the condenser outlet pipe shown in Fig. 18.4. The condenser had been off line for cleaning. The rat, having crawled up the riser pipe to the reflux drum, got its head stuck in the drum's inlet nozzle. Your author, unaware of the rodent's predicament, put the exchanger back into service. The condensed butane now flowed across the rat. The rat died. Well, we all must come to that end eventually, although perhaps not quite that exact end. Such is the way of all flesh.

This rat is called a "20-lb rat." Not that the rat weighed 20 lb. The 20 lb refers to the pressure drop of 20 psi that the liquid encountered as it flowed across the rat's now-lifeless body. Before the introduction of this pressure restriction, the butane entering the reflux drum was at its bubble point. Our question is, will the introduction of the rat at the inlet nozzle cause the butane, as it enters the vessel, to flash?

The term "to flash" is used to denote partial vaporization of the butane. Before the rat became stuck, the liquid entering the reflux drum did not flash or partly vaporize.

We can also be sure that at steady state the butane liquid did not flash after the introduction of the rat because no vapor was vented from the reflux drum.

But let's assume that in the first microsecond after the introduction of the rat the liquid did vaporize. The vapor so generated would

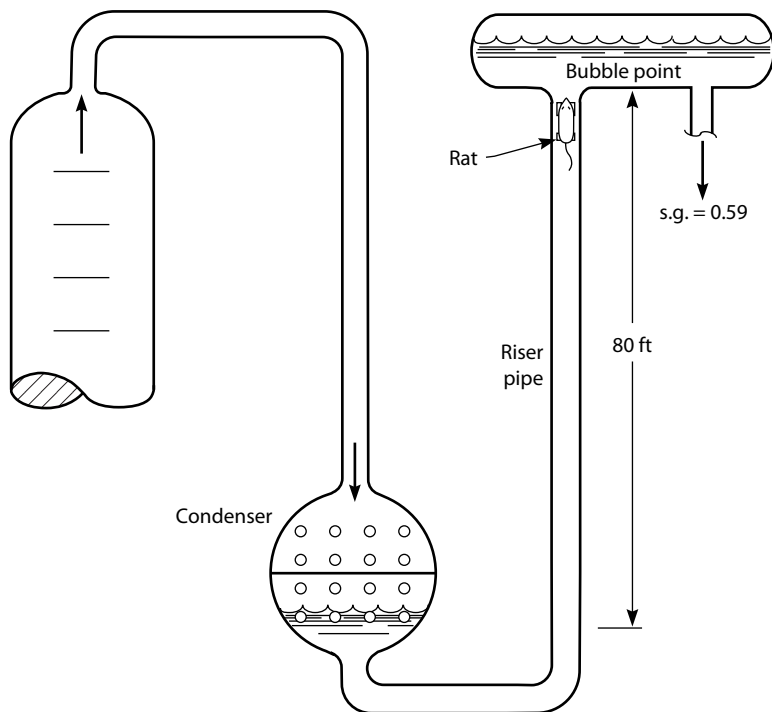


FIGURE 18.4 Elevation increase of the reflux drum increases tower pressure.

be trapped in the reflux drum. The pressure in the drum would increase. Not by 20 psig, but just a little. The small increase in pressure in the reflux drum would push up the liquid level in the condenser. The surface area of the condenser available to subcool the liquid would increase. The liquid temperature would be reduced. As the subcooled liquid flowed across the dead rat, its pressure would drop. The liquid's pressure would fall to exactly that pressure that corresponds to the vapor pressure of the butane at the temperature in the reflux drum.

The backup of butane liquid in the condenser would continue until the butane leaving the condenser was cold enough so that it would not flash as it flowed across the rat—that is, until *equilibrium conditions* had been re-established in the reflux drum.

As the butane liquid level in the condenser increased, the area of the exchanger exposed to the condensing vapors would decrease. Let's assume that the tower's reboiler duty was constant. The vapor flow rate to the condenser would then be constant. To condense the same flow rate of vapor with a shrinking exchanger surface area, the pressure of condensation must increase. The tower pressure would also go up as the condenser pressure rose.

Forget about the rat. I made that story up. The real story is that the riser pipe connecting the condenser outlet to the reflux drum was undersized. Nat Taylor, the project engineer, specified a 4-inch pipe when an 8-inch pipe was needed. The pressure drop through the pipe was then 32 times higher than the intended ΔP of 0.60 psig (ΔP in a pipe varies to the fifth power of the pipe's diameter). The resulting riser pipe pressure drop was 20 psi. This *frictional loss* of 20 psi in the pipe had the exact same effect on the condenser—and on the tower's pressure—as the 20-lb rat.

Forget about Nat. Forget about the rat. Both stories are pure fiction. The truth is that the elevation of the reflux drum was 80 ft above the condenser. The specific gravity of the butane liquid was 0.59. This means that 80 ft of liquid exerted a head pressure of about 20 psig:

$$\frac{80 \text{ ft} / 2.31 \text{ ft} \cdot \text{H}_2\text{O}}{\text{psi}} \times \frac{0.59 \text{ s.g. butane}}{1.00 \text{ s.g. H}_2\text{O}} = 20 \text{ psig}$$

where s.g. = specific gravity.

This *elevation head loss* of 20 psig had the same effect on the condenser—and on the tower's pressure—as did the rat or Nat's frictional loss.

18.1.3.5 Common Design Error

Please refer back to Fig. 18.2. How can the liquid from the condenser rise to the higher elevation in the reflux drum without being pumped? Simple! The pressure head of the liquid leaving the condenser is converted to elevation as the liquid flows up into the reflux drum. This works fine as long as the liquid leaving the condenser is sufficiently subcooled. By "sufficiently subcooled," I mean that when the lower-pressure liquid flows into the reflux drum, it has to be cold enough so that it does not flash.

The liquid leaving the condenser is subcooled. The liquid entering the reflux drum is saturated liquid at its bubble point. Of course, the temperature of the liquid is the same at both points. The subcooled liquid is "subcooled" in the sense that its pressure is above the bubble-point pressure at the condenser outlet temperature. It is this extra pressure, above the bubble-point pressure, that may be converted to elevation.

I have another way to explain how the condensed liquid can flow uphill from the condenser into the elevated reflux drum. That is, the liquid leaving the condenser is pushed into the reflux drum by the higher pressure in the tower. This will work okay, as long as the liquid entering the reflux drum is at its saturated bubble-point temperature and pressure. If this is not the case, then the vapors flashing out in the reflux drum will have to be vented. If you refuse to open this vent, then condensate backup must increase in the condenser, with a consequent reduction in the condenser's capacity.

18.2 Pressure Control

18.2.1 Tower Pressure Control

For total condensers, there are three general schemes for controlling distillation tower pressure:

- Throttling the cooling water flow to the condenser
- Flooding the condenser
- Hot-vapor bypass around the condenser

Regardless of the method selected, the principal concept of tower pressure control is the same. We control the pressure in the reflux drum by manipulating the temperature in the reflux drum. The tower pressure then floats on the reflux drum pressure. To lower the tower pressure, we must first cool the reflux drum. This reduces the vapor pressure of the liquid in the reflux drum.

The oldest, most direct method of pressure control is throttling on the cooling-water supply. This scheme is shown in Fig. 18.5. Closing the water valve to the tube side of the condenser increases the condenser outlet temperature. This makes the reflux drum hotter. The hotter liquid in the reflux drum creates a higher vapor pressure. The higher pressure in the reflux drum increases the pressure in the tower. The tower pressure is the pressure in the reflux drum plus the pressure drop through the condenser.

Throttling on the cooling water works fine, as far as pressure control is concerned. But if the water flow is restricted too much, the cooling-water outlet temperature may exceed 125 to 135°F. In this temperature range, water-hardness deposits plate out inside the tubes. Then the heat-transfer coefficient is permanently reduced by the fouling deposits.

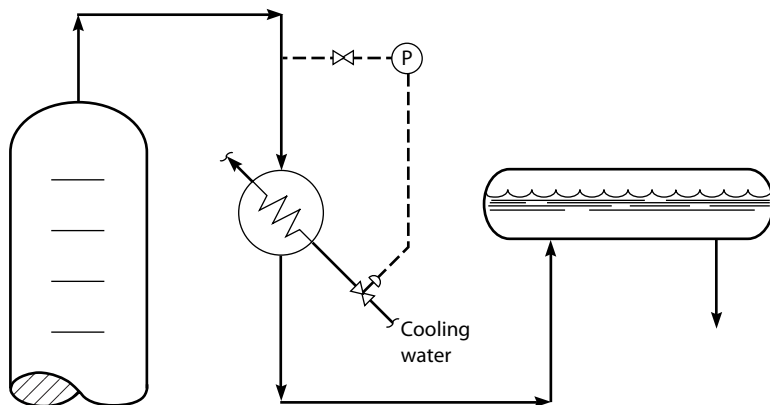


FIGURE 18.5 Tower pressure control using cooling-water throttling.

18.2.2 Hot-Vapor Bypass

18.2.2.1 Hot-Vapor Bypass Pressure Control

A more modern way of controlling a tower's pressure is shown in Fig. 18.6. This is the hot-vapor bypass method. When the control valve on the vapor bypass line opens, hot vapors flow directly into the reflux drum. These vapors are now bypassing the condenser. The hot vapors must condense in the reflux drum. This is because there are no vapors vented from the reflux drum. So, at equilibrium, the hot vapors must condense to a liquid on entering the reflux drum. They have no other place to go.

The latent heat of condensation of this vapor is absorbed by the liquid entering the reflux drum. The liquid that enters the reflux drum comes from the condenser. The hot vapor mixes with the condenser outlet liquid and is condensed by this cooler liquid.

Does this mean, then, that the condenser outlet temperature is lower than the reflux drum outlet temperature? If I were to place my hand on the reflux pump suction line, would it be warmer than the condenser outlet? Answer—yes!

But the liquid in the reflux drum is in equilibrium with a vapor space. This liquid is at its bubble, or boiling point. If the liquid draining from the condenser is colder than this bubble-point liquid, it must

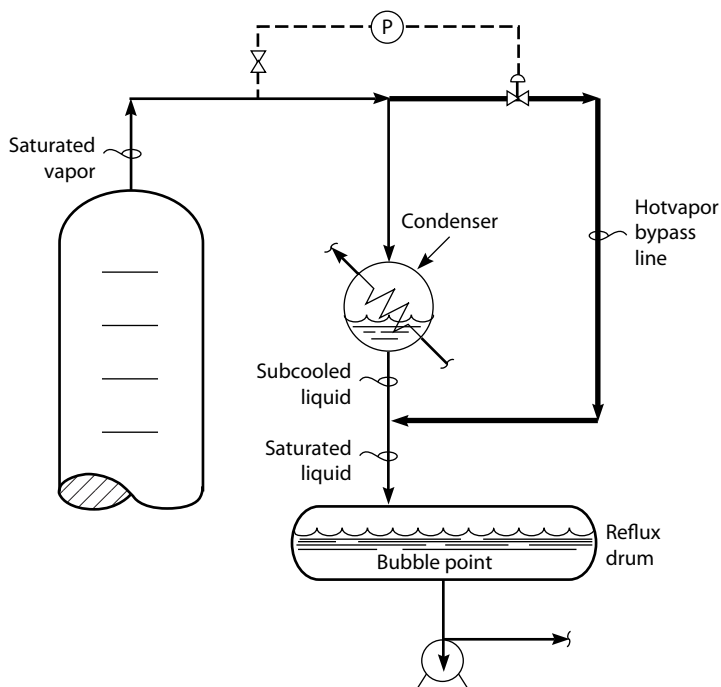


FIGURE 18.6 Hot-vapor bypass pressure control. Note condensate backup.

be subcooled. But how can a vapor condense directly into a subcooled liquid? Well, it cannot.

The tower overhead vapor, shown in Fig. 18.6, condenses to a liquid on the outside of the cold condenser tubes. The liquid drips off the tubes. These droplets of liquid are in close contact with the saturated vapor in the condenser shell. This means that the liquid is in equilibrium with the vapor. The condensed liquid is therefore initially at its bubble-point temperature. This liquid accumulates in the bottom of the condenser's shell. The submerged tubes then must subcool this liquid. Part of the surface area of the condenser is hence devoted to subcooling liquid, and part is devoted to condensing vapor.

But how does this work? How does the condenser know, without any advice from us, how much of its heat-exchanger surface area, which is supposed to be used to condense vapors, to divert to subcooling liquid?

Let's assume that the liquid draining from the condenser is not quite cold enough to absorb the entire latent heat of condensation of the vapors flowing through the hot-vapor bypass line. The vapors will then be only partially condensed. Vapor will start to accumulate in the reflux drum. This accumulation of vapor will increase the reflux drum pressure by a small amount. The higher drum pressure will back up the liquid level in the condenser by a few inches. The higher height of liquid in the drum will submerge additional cold tubes with the condensed liquid. The cooler liquid will now be able to absorb more of the latent heat of condensation of the vapor passing through the hot-vapor bypass line. Eventually, a new equilibrium will be established.

18.2.2.2 Leaking Hot-Vapor Bypass Valve

Let's assume that the hot-vapor bypass valve, shown in Fig. 18.7, is leaking. It is leaking 10 percent of the tower overhead flow. A good rule of thumb is then

- *Hydrocarbons.* For each 20°F temperature difference between the cooler condenser outlet and the warmer reflux pump suction, 10 percent of the tower's overhead vapor flow is leaking through the hot-vapor bypass valve.
- *Aqueous systems.* For each 20°F temperature difference between the condenser outlet and the reflux pump suction, 1 percent of the tower's vapor flow is leaking through the vapor bypass valve.

As the hot-vapor bypass valve opens, the condensate level in the shell side of the condenser increases to produce cooler, subcooled liquid. This reduces the surface area of the condenser exposed to the saturated vapor. To condense this vapor with a smaller heat-transfer area, the pressure of condensation must increase. This, in turn, raises the tower pressure. This, then, is how opening the hot-vapor bypass pressure-control valve increases the tower pressure.

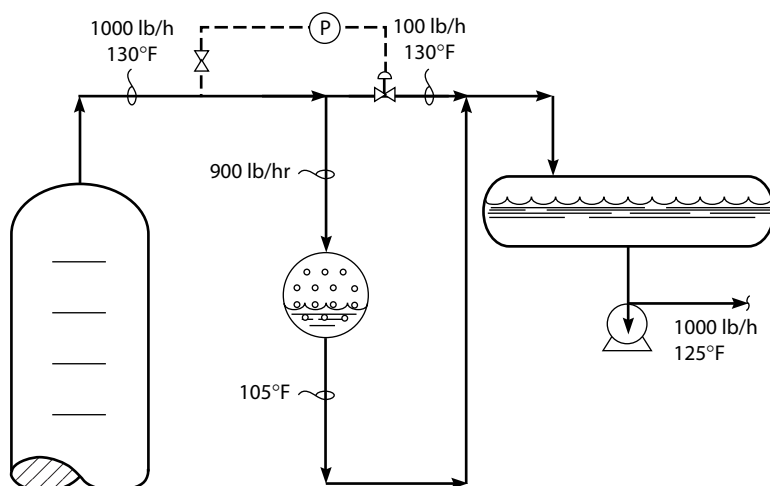


FIGURE 18.7 Opening pressure-control valve cools condenser outlet.

Incidentally, as I have shown in Figs. 18.6 and 18.7, the condenser may be located above or below the reflux drum. Both configurations require a subcooled liquid effluent from the condenser. But if the condenser is located below the reflux drum, additional subcooling to offset the elevation effect, described above, will be needed.

18.2.3 Flooded Condenser Pressure Control

You should have concluded by now that hot-vapor bypass pressure control actually works by varying the surface area of the condenser exposed to the saturated vapor. But why do this indirectly? Why don't we simply and directly vary the liquid level in the condenser, as shown in Fig. 18.8?

The answer has something to do with women's clothes. Perfectly good, sensible designs go out of fashion. The perfectly good, sensible design used to control a tower's pressure, *flooded condenser control*, shown in Fig. 18.8, has gone out of fashion. In a flooded condenser tower pressure-control strategy, the reflux drum is run full. Restricting the flow from the reflux pump increases the level in the condenser. This reduces the heat-transfer surface area available for condensation and raises the tower pressure. Either the reflux or overhead product flows may be used to vary the liquid level in the condenser.

Once a liquid level reappears in the reflux drum, the condenser capacity has been exceeded. The level in the condenser will continue falling until the drum empties and the reflux pump begins to *cavitate*.

In general, flooded condenser pressure control is the preferred method to control a tower's pressure. This is so because it is simpler and cheaper than hot-vapor bypass pressure control. Also, the

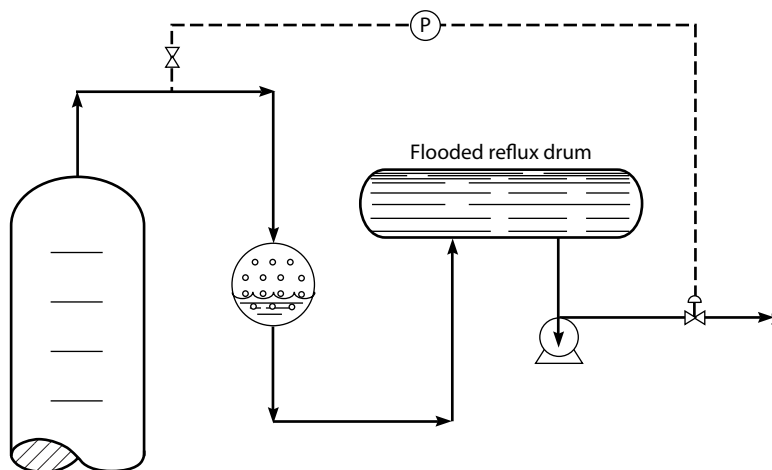


FIGURE 18.8 Flooded condenser pressure control: the preferred method.

potential problem of a leaking hot-vapor bypass control valve cannot occur. Many thousands of hot-vapor bypass designs have eventually been converted—at no cost—to flooded condenser pressure control.

The function of the reflux drum in a flooded condenser design is to

- Separate water from reflux when distilling hydrocarbons
- Give the operators time to respond if they have exceeded the condenser's capacity
- Provide a place from which non-condensable vapors may be vented

One problem with flooded condenser pressure control is related to the need to occasionally vent non-condensables. This vent valve must not leak when closed. Since the drum is normally full of liquid, a leaking vent valve will pass liquid. Many pounds of product can be lost in this way. A butterfly control valve with a soft, rubberized seat is a good choice for a remotely controlled, non-condensable vent valve.

Air-cooled condensers seem to work as well as shell-and-tube water coolers in flooded condenser pressure control service. Some air coolers are sloped toward the outlet to reduce the ratio of the delta condenser surface area to delta height change of condensate backup. I don't think this is necessary, but it may help in fine-tuning pressure control with air-cooled flooded condensers.

But don't just take my word for it. Try running on flooded condenser pressure control rather than hot-vapor bypass, and I think you will be quite pleased with the result.

18.2.4 Partial Condensation

If we normally have a situation in which non-condensable vapors appear in the reflux drum, then there is only one pressure-control option available. This is to place the tower pressure-control valve on the vapor off-gas as shown in Fig. 18.9. If we normally have non-condensable vapors in the condenser effluent, then the following problems we have been discussing do not exist:

- Condensate backup
- Subcooling of condenser effluent
- Fouling due to low flow of the cooling-water tubes

Sometimes we see tower pressure control based on feeding a small amount of inert or natural gas into the reflux drum. This is bad. The natural gas dissolves in the overhead liquid product and typically flashes out of the product storage tanks. The correct way to control tower pressure in the absence of non-condensable vapors is to employ flooded condenser pressure control. If, for some external reason, a variable level in the reflux drum is required, then the correct design for tower pressure control is a hot-vapor bypass.

18.2.5 Slug Flow in Risers

Figure 18.10 illustrates a common cause of an erratic tower operating pressure. The condenser effluent is a two-phase mixture of vapor and liquid. In the summer, the tower pressure is steady. But in the winter when the cooling water is cold, pressure control is erratic.

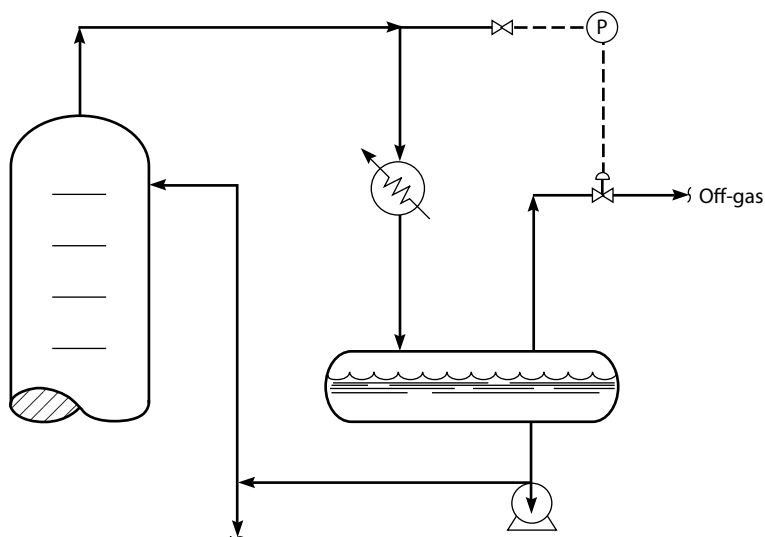


FIGURE 18.9 Pressure control for partial condensation.

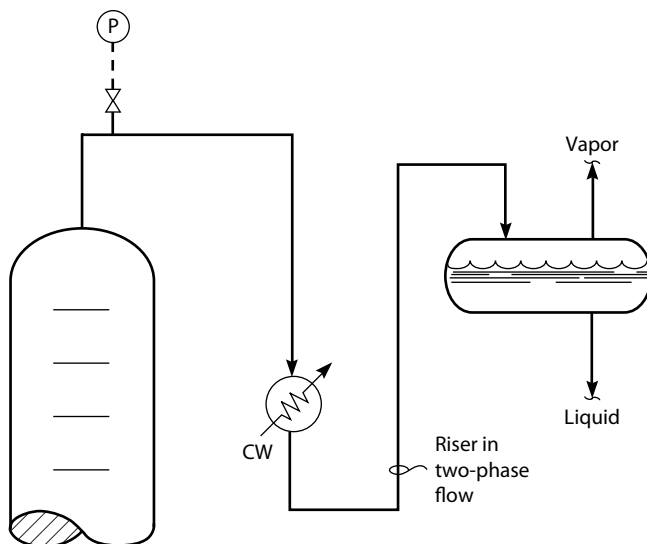


FIGURE 18.10 Slug flow in risers.

The problem is phase separation in the riser line. The vertical velocity in this line is too low to maintain a froth flow up the riser. Slugs of liquid form. Periodically the accumulating liquid is blown out by the vapor. This slug flow causes an erratic back pressure on the tower but not an erratic pressure in the drum.

This problem only occurs when the drum is elevated above the partial condenser. The fundamental difficulty is excessive riser pipe diameter. To avoid slug flow in such a riser a mixed-phase velocity of at least 20 ft/s is needed.

While the 20 ft/sec may be maintained in the summer, more efficient condensation in the winter may reduce vapor flow. This can cause the riser velocity to drop below the minimum to prevent phase separation. Throttling the cooling water will stabilize the tower pressure, but may result in salting up the exchanger with water-hardness deposits.

CHAPTER 19

Air Coolers

Fin-Fan Coolers

Air coolers are twice as expensive to purchase and install as water coolers. The great advantage of an air cooler is that it does not need cooling water. The difficult aspect of air cooling arises from the flow of air across the tubes.

Most air coolers are either induced-draft or forced-draft, as shown in Fig. 19.1, the more common arrangement being forced-draft. The air is moved by rather large fans. The tubes are surrounded with foil-type fins, typically 1 inch high. The surface area of the fins as compared to the surface area of the tubes is typically 12 to 1. That is why we call an air cooler an *extended-surface* heat exchanger.

The heat-transfer coefficient of an air cooler (Btu, per hour, per square foot of finned area, per degree Fahrenheit) is not particularly good. It might be 3 to 4 for cooling a viscous liquid, or 10 to 12 for condensing a clean vapor. The low heat-transfer coefficients are offset by the large extended surface area.

19.1 Fin Fouling

In a forced-draft air cooler, cool air is blown through the underside of the *fin tube bundle*. In an induced-draft air cooler, cool air is drawn through the underside of the fin tubes. Either way, road dust, dead moths, catalyst fines, and greasy dirt accumulate along the lower row of tubes. As the tubes foul, they offer more resistance to the air flow. However, note that for forced-draft fans:

- The total air flow discharged by the fan remains constant regardless of the fin tube fouling.
- The fan discharge pressure remains constant regardless of the fin tube fouling.
- The amperage electric load on the motor driving the fan remains constant regardless of the fin tube fouling.

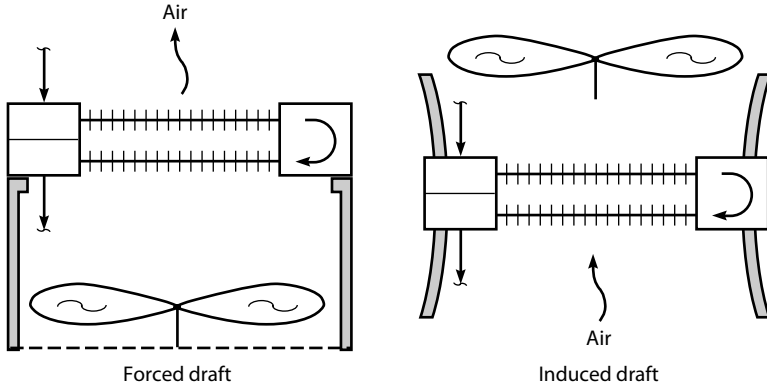


FIGURE 19.1 Two types of air coolers.

Figure 19.2 explains this apparent contradiction. As the underside of the fins becomes encrusted with dirt, an increasing amount of air is reflected back through the screen, located below the fan. The air is reflected back through the screen in a predictable pattern. The air flow in the center of the screen is always going up, which is the desired direction of flow. The air flow around the edge of the screen is always reversed, which is the wrong direction.

As the exterior fouling on the tubes worsens, the portion of the screen through which the air flows backward increases. As the dirt accumulates on the underside of the tubes, the portion of the screen through which the air is drawn upward decreases. Even though the air flow blown through the bundle is decreasing, the total air flow delivered by the fan is constant.

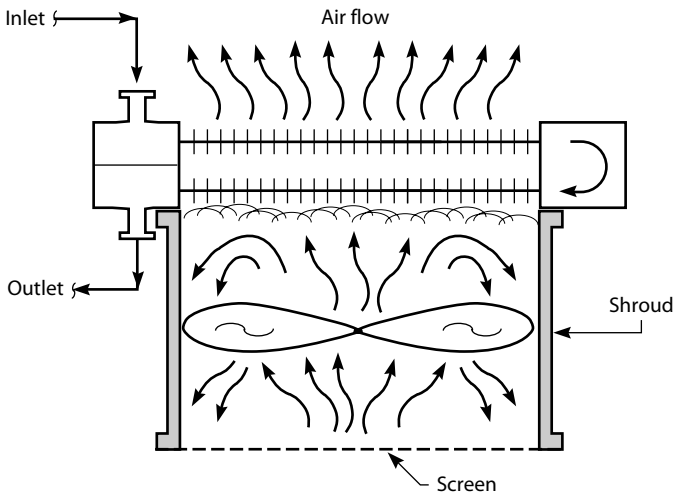


FIGURE 19.2 Air flow under partially plugged bundle.

19.2 Fan Discharge Pressure

Fan operation is indicated on a performance curve, as shown in Fig. 19.3. The head developed by the fan is equivalent to 5 or 10 inch of water. As the fan air flow is pretty constant, the fan's head is also constant. Another way of stating this is to say that as a tube bundle fouls, the resistance to air flow increases. This reduces the air flow through the bundle, but the pressure loss of the air flow through the tube bundle does not change.

If the head developed and the flow produced by a fan are both constant, then the power needed to run the fan must also be constant. Why? Because the power needed to spin a fan is proportional to the produced flow and the produced head.

To prove this to yourself, find the electric circuit breaker for a fan's motor. The amp (amperage) meter on the circuit breaker will have a black needle and a red needle. The black needle indicates the actual current, or amp load. The red needle is the amperage load that will trip the motor as a result of *overamping*. Over time, as the tube bundle fouls and air flow through the bundle of a forced-draft fan is restricted, the black needle never moves.

An induced-draft fan (see Fig. 19.1) is a different story. As the tube bundle fouls,

- The air pressure to the fan drops.
- The air pressure from the discharge of the fan is just atmospheric pressure. It remains constant.
- The water head (in inches) developed by the fan increases.
- The flow of air through the fan and the bundle decreases. This is consistent with Fig. 19.3.
- The amp load on the motor spinning the fan decreases.

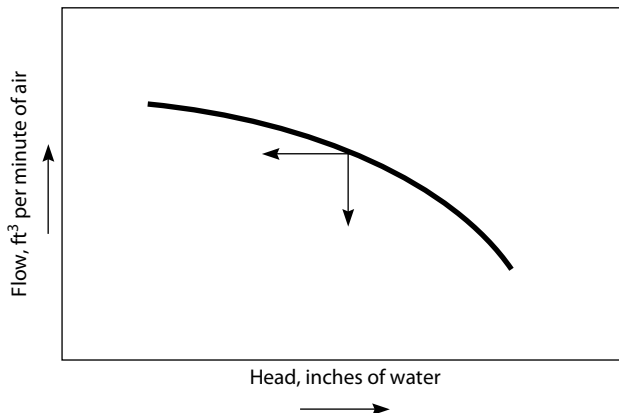


FIGURE 19.3 Fan performance curve.

Naturally, there is no reverse air flow on an induced-draft fan. That can occur only in a forced-draft fan. Reverse air flow can be observed with a forced-draft fan by seeing which portions of the screen, shown in Fig. 19.2, will not allow a dollar bill to stick to the underside of the screen.

19.3 Effect of Reduced Air Flow

Loss of air flow through a finned tube air cooler bundle is a universal problem. The effect is to reduce the exchanger's cooling efficiency. To restore cooling, you might wish to try the "Norm Lieberman method," which consists of reversing the polarity of the three-phase fan motor electric leads. The fan will now spin backward. Depending on the nature of the deposits, a portion of the accumulated dirt will be blown off the tubes—but all over the unit. Personnel should observe this procedure from a "safe" distance.

A more socially acceptable option is to water-wash the tubes. Most of the effective washing must be underneath the tubes. Washing from the top down is relatively ineffective. In many cases, detergent must be added to the washwater to remove greasy dirt. (*Caution:* Hot tubes may be thermally shocked by this washing and pull out of the tube header box.)

To effectively water-wash the deposits from the fins I will proceed as follows:

1. Shut off the fan.
2. Lock out the motor.
3. Tie off the fan blade with a rope to keep it from spinning in the wind.
4. Remove the screen.
5. Use a 0.5-inch piece of tubing as a washing wand.
6. For water, use steam condensate or boiler feed water at 50 to 100 psig.
7. Hold the wand tip 6 to 10 inches from the bottom row of tubes.
8. Run the wand between each row of tubes individually.

This is a long, wet, hot, and dirty job, but the results are sometimes quite fantastic. Washing from the top down takes even longer, uses 10 times as much water, and never does a good job. But it's a lot easier and drier.

The fan blades themselves may be adjusted to obtain more air flow. This is done by increasing the *fan blade pitch*. The pitch can usually be adjusted between 12° (for low air flow) to 24° (for high air flow). Any increase in air flow has to increase the amp load on the fan motor driver. The motor could then trip off.

Setting the blade pitch cannot be done with great precision, and it's not too critical. I once increased the blade pitch from 15° to about 22°. Air flow increased by only 5 percent measured by the increased amperage load on the motor driver.

Cooler weather always increases the air flow produced by a fan. This always increases the amp load on the fan's motor driver. To prevent the motor from tripping off, or simply to save electricity during the winter, you might reduce the fan blade pitch.

One factor that does not reduce air flow is crushed fins at the top of the tube bundle. Walking across a fin tube bundle will crush these fins. It looks bad, but does not appear to affect cooling efficiency.

Take a close look at Fig. 19.2. Note that on the right side of the sketch there is a small gap between the blade tip and the shroud. It is this gap that accounts for the air recirculation previously described. The bigger the gap, the greater the detrimental air recirculation. With age, shrouds get out of round and the gap increases, but not uniformly.

The only way to seal off this gap is to use strips of plastic or Teflon attached to the inner wall of the shroud. When the fan is turned on it will cut through parts of the plastic strips and create its own seal. Field results have been positive, and the strips, called "tip seals," can be purchased as a retrofit kit from air cooler vendors.

19.4 Adjustments and Corrections to Improve Cooling

19.4.1 Adjusting Fan Speed

The revolutions per minute (rpm) (or rotational speed) of a fan can be increased by increasing the size of the *motor pulley*, which is the grooved wheel on the motor shaft. A small increase in the diameter of this pulley will greatly increase air flow through the cooling bundle. But according to the *affinity* or *fan laws*, doubling the diameter of a pulley increases the driver amp load by 800 percent. That is, driver horsepower increases to the cube (third power) of the fan's speed.

But there is a bigger problem than motor overload when increasing a fan's speed. The blades themselves are rated only for a maximum centrifugal force. This force increases with increased fan rpm. At some maximum speed, the blades fly apart. Gentle reader, you can imagine how I became so smart on this subject.

Belt slipping used to be a major problem on air coolers. The resulting low rpm routinely reduced air flow. Modern air coolers have notched belts, which are far less subject to belt slippage. Regardless, a slipping belt will result in a reduced amp load on the fan's motor driver.

19.4.2 Use of Water Sprays on Air Coolers

Spraying water on fin-fan air coolers is generally not a good idea. It is really effective only in dry climates with low humidity. The evaporation of water by the dry air cools the surface of the fins; that is, the latent heat of vaporization of the water robs sensible heat from the tubes.

Salts or other dissolved solids in the evaporating water will plate out on the exterior of the tubes. With time, a serious loss in heat-transfer efficiency results. Use of steam condensate can avoid this particular difficulty.

Water sprays should be used only as a stopgap measure because of the swell they cause in the plant's effluent volume, and also their tendency to create a safety hazard in the vicinity of the cooler.

One of my clients used fire water for a few hours to cool an air cooler. The problem was the fire water was seawater. It proved impossible to totally remove the salts from between the fins. The tube bundles had to be replaced to restore efficiency.

19.4.2.1 Sweeping the Underside of an Air Cooler Tube Bundle

A finned tube bundle acts as an air filter. Air is drawn up through the lower two rows of fins where much of the dust, dirt, bugs, and moths are filtered out, and thus restricts air flow. Water-washing is the most effective method of removing these deposits. However, many of my clients will not water-wash hot air cooler bundles while running for fear of pulling out the tubes from the tube sheets due to thermal contraction of the tubes.

A quick, but rather dirty method that I've used to remove many of these deposits is to drop off the screen, turn off the fan, then lock out the fan and tie off the fan with a strong rope, and with a long-handled broom, sweep off the underside of the finned tubes. This rather primitive technique is sometimes surprisingly effective.

19.4.2.2 Fin Deterioration

The fins are usually made out of aluminum. Especially in moist, steamy environments, the fins are subject to destruction by corrosion. A corroded fin retards rather than promotes heat transfer. It's easy to break such fins off by hand. A high-pressure jet of water can be used to knock off the corroded fins and partly restore cooling capacity.

19.4.3 Air Cooler Fan Alignment

For forced-draft fans, I have sometimes seen that the fan assembly gets slightly out of true vertical alignment. One side of the fan is running a little higher than the other. The consequences of this misalignment will be to increase the clearance between the fan blade tip and the inside of the shroud by perhaps a few inches. This increased blade-to-shroud clearance will promote air recirculation from the discharge of the fan back to the bottom of the shroud. The net result is a reduction in the air flow through the finned tube bundle.

The problem can be corrected by realigning the fan assembly. Alternately, vane tip clearance mesh can be attached to the interior of the shroud, which will eliminate excessive blade-to-shroud clearance and thus push more cooling air through the finned tube bundle.

Alternately, the shroud itself may become distorted with age. This will also cause the clearance between the blade tips and the inside of the shroud to increase. This vane tip clearance mesh is also used to correct this problem, which promotes air recirculation and diminishes air flow through the bundle.

I've only used this mesh once. It came in a big roll several feet high and about 8 inches wide. The mesh itself was around 2 inches thick and screwed into the inside of the shroud. A simple tool was then bolted onto one blade. After that the fan was turned by hand to cut a groove into the mesh.

It seemed to me that air flow increased by some amount that I did not measure. The operators reported that cooling had been increased.

19.5 Designing for Efficiency

19.5.1 Tube-Side Construction

The mechanical construction of the tubes in an air cooler creates some rather nasty problems. Figure 19.4 shows the exterior appearance at either end of an air cooler. The small black circles are threaded steel plugs. They are not connected to the ends of the tubes. Allow me to rotate the air-cooler header box shown in Fig. 19.4 by 90° and display a cross-sectional view in Fig. 19.5. Note that the plugs are not connected to individual tubes. Unscrewing a plug just gives one access to the end of a tube for cleaning purposes.

Proper cleaning of an air-cooler tube requires removing two plugs. A large industrial air cooler may have 2000 tubes or 4000 plugs. The labor involved to remove and reinstall all these plugs is formidable.

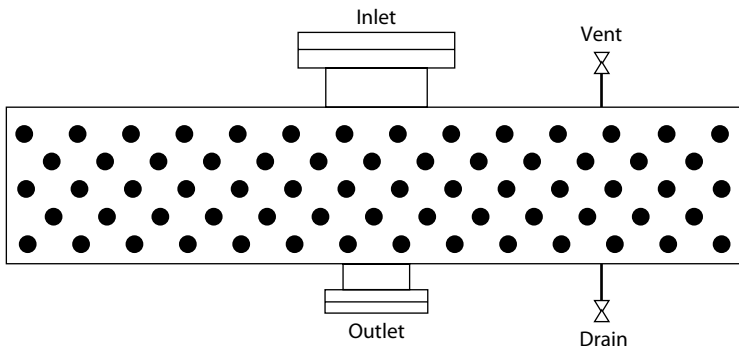


FIGURE 19.4 End view of an air-cooler header box.

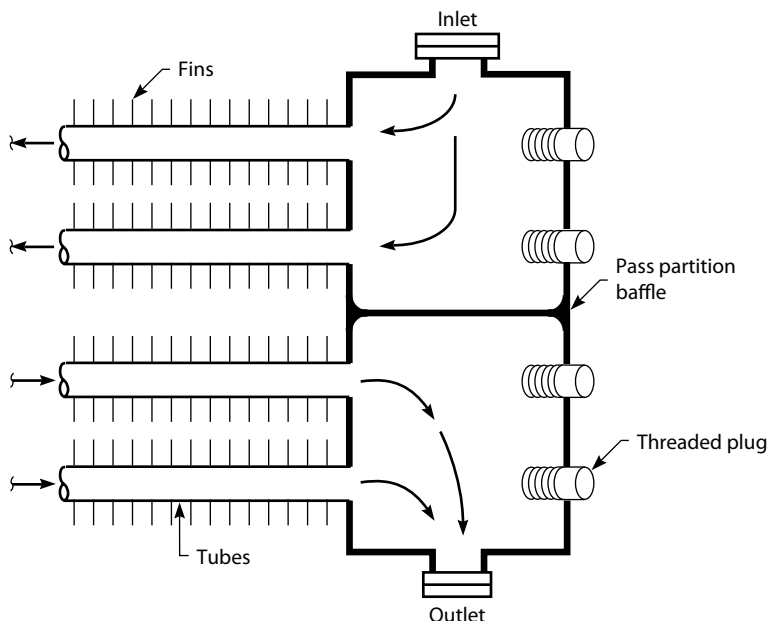


FIGURE 19.5 Cross-section of an air-cooler header box.

Leaking plugs due to cross-threading is a common start-up problem. Hence, many air coolers are simply never cleaned.

The pass partition baffle shown in Fig. 19.5 makes this cooler a two-pass exchanger. These baffles are subject to failure due to corrosion. More often, they break because of excessive tube-side pressure drop. The differential pressure across a two-pass pass partition baffle equals the tube-side ΔP .

Once the pass partition baffle fails, the process fluid may bypass the finned tubes, and cooling efficiency is greatly reduced. This is bad. But worse yet, during a turnaround of the cooler, there is normally no way to inspect the pass partition baffle. There is no easy way to visually verify the mechanical integrity of this baffle. A few air coolers have removable inspection ports for this purpose; most do not.

19.5.2 Parallel Air Coolers

A large process plant air cooler may have 10, 20, 30, or more banks of air coolers, arranged in parallel. Figure 19.6 shows such an arrangement. Let's assume that the inlet header is oversized and has zero pressure drop. Let's also assume that the outlet header is oversized and also has no ΔP . The pressure drop across the tube side of all such air coolers arranged in parallel is then identical.

If one of the air coolers begins to experience tube-side fouling, the fluid flow will be reduced. But the tube-side pressure drop will remain

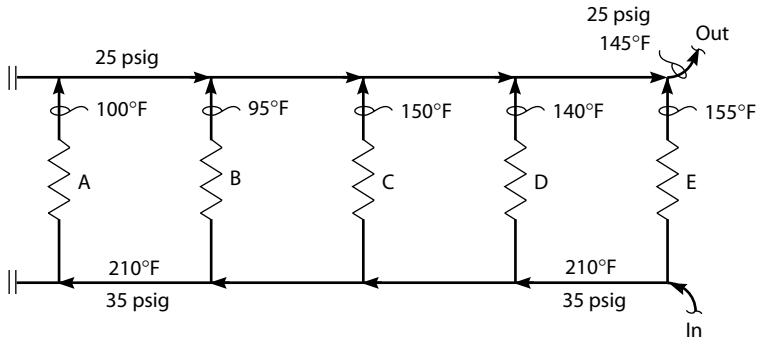


FIGURE 19.6 Air-cooler banks in parallel.

the same. The pressure drop across all five air-cooler bundles, shown in Fig. 19.6, is 10 psig.

Individual flows to parallel banks of air coolers are rarely—if ever—measured. Regardless, we can gauge the approximate relative flow to each bundle. This can be done by checking the outlet temperature of the bundles or banks.

Let's assume that the cooling air flow to all five banks is the same. Banks A and B in Fig. 19.6 have low outlet temperatures. Banks C, D, and E have much hotter outlets. Question: Which coolers are handling most of the heat-transfer duty? Is it A and B or C, D, and E?

The correct answer is C, D, and E. Most of the flow is passing through C, D, and E. Very little flow is passing through A and B. Look at the combined outlet temperature from all five coolers. It is 145°F. This indicates that most of the total flow is coming from C, D, and E—the banks with the higher outlet temperature. Very little of the flow is coming from A and B—the banks with the lower outlet temperature.

Why would the flow through A and B be so low? Apparently, their tubes must be partly plugged. Corrosion products, gums, and dust are common plugging agents. But when such exchangers foul, their relative tube-side ΔP , as compared to the other exchangers, remains constant. But their relative tube-side flow, as compared to the other parallel exchangers, decreases.

19.5.2.1 Air Outlet Temperature

The individual air outlet temperatures from the coolers shown in Fig. 19.6 are

- A: 105°F
- B: 95°F
- C: 170°F
- D: 165°F
- E: 180°F

These temperatures may be measured with a long-stem (24-inches) portable temperature probe. Do not touch the tip of the probe to the fins when making a reading. Four readings per tube bundle section are adequate to obtain a good average.

The ambient temperature was 85°F. The individual temperature rises for each air cooler would then be

A =	105°F – 85°F =	20°F
B =	95°F – 85°F =	10°F
C =	170°F – 85°F =	85°F
D =	165°F – 85°F =	80°F
E =	180°F – 85°F =	95°F
<hr/>		
Total	=	290°F

If you are now willing to make the assumption that the air flow is the same through the five coolers, we could calculate the process side flow through each cooler. For example, percent flow through A = 20°F/290°F = 7 percent. This calculation assumes that the percent of flow through the cooler is proportional to the air temperature rise through the cooler divided by the total air temperature rise through all five coolers.

It is not all that difficult to decide whether the air flow through identical coolers is similar. I just wave a handkerchief in the breeze at a few spots above the cooler.

19.5.3 Air-Cooled Condensers

In many process plants, the pump alleys are covered with forced-draft, air-cooled condensers. Dozens of coolers are arranged in parallel. I have seen services where 300 mm Btu/h of condensation duty was easily handled by aerial cooling. All these large systems had one problem in common. They all tended to have higher flows through cooling banks connected closest to the inlet and/or outlet headers. The higher relative flows were indicated by both higher air outlet and higher process outlet temperatures. A good example of this is shown in Fig. 19.6.

The mechanism that causes this often-severe flow maldistribution is based on low-temperature dew-point corrosion, as explained below. Here is a rather common example, assuming that the main corrodants are chlorides, in a hydrogen sulfide-rich, condensing hydrocarbon vapor:

1. A small amount of ammonia is injected into the overhead vapor line to control the pH of a downstream water draw.
2. Vapor-phase ammonium chloride is formed.
3. Tube bundles nearest the inlet header tend to see perhaps 1 percent more flow than the tube bundles farthest from the inlet header.

4. Bundles seeing the lower flows have slightly lower outlet temperatures.
5. Lower temperature favors the *sublimation* of ammonium chloride vapor to a white saltlike solid.
6. This salt is very *hygroscopic*, meaning that it will absorb and condense water vapor from the flowing hydrocarbon vapor stream at unexpectedly high temperatures.
7. The resulting wet chloride salts are very corrosive, especially to carbon steel tubes.
8. A ferric chloride corrosion product is formed.
9. This metallic chloride salt then reacts with the abundant molecules of hydrogen sulfide to produce hydrochloric acid and iron sulfide.
10. The hydrochloric acid may then continue to promote corrosion.
11. The iron sulfide (or pyrophoric iron) accumulates as a blackish-gray deposit inside the tubes.
12. This deposit further restricts vapor flow through the low-flow tubes.
13. The reduced flow causes a lower tube outlet temperature.
14. The lower tube outlet temperature promotes higher rates of salt sublimation from vapor to a corrosive fouling solid.

Meanwhile, the air-cooler bundles nearest the inlet header tend to see a greater and greater percentage of the total flow as the cooler bundles foul and plug. They tend to stay hot and clean, and those bundles farthest from the inlet header tend to run cool and dirty.

It is a general principle of heat exchange that low flows tend to promote fouling and fouling promotes corrosion. The corroded, fouled heat-exchanger surface retards flow and creates a vicious cycle. We will see this problem again in shell-and-tube heat exchangers, as discussed in Chap. 26.

The best way to handle the nonsymmetrical flow problem described previously is to make the pressure drops in both the inlet and outlet tube bundle headers very small, as compared to the bundle pressure drop itself. Many of my clients add additional tube bundles in parallel with existing air coolers. This helps at first, but they find that the long-term benefits are quite disappointing because of high pressure drop in the new header lines.

Slug washing individual tube bundles with steam condensate also helps. Washings for 20 minutes once a week are often sufficient.

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CHAPTER 20

Thermodynamics

How It Applies to Process Equipment

While most readers of this text will have studied thermodynamics at university or while obtaining their steam ticket, few have understood it. That's because it was not explained properly. I can and will rectify this situation in this easy-to-read short chapter.

I will mainly be discussing water and steam. This is the working fluid that we on planet Earth have selected to run many of our machines and generate most of our electricity. The word *thermo* refers to the heat content of steam. The word *dynamics* refers to the velocity of steam:

- Thermo = Heat
- Dynamics = Velocity

20.1 Why Is Thermodynamics Important to the Plant Operator?

Machines are not driven by the heat of steam, but by the velocity of steam. Mainly, I'm referring to steam turbines used to drive pumps, compressors, air blowers, and electrical generators. It's the velocity of the steam striking the turbine blades that causes the turbine to spin. The greater the velocity of the steam striking the turbine blades, the more work can be extracted from each pound of steam. The application of thermodynamics in the process plant is intended to maximize the velocity of steam.

However, the source of steam velocity does not originate with the pressure of the steam, but with the heat content of the steam. Hence, the original text on thermodynamics was called *Heat, a Mode of Motion*. The term *thermodynamics* was introduced as an afterthought because it sounds more important.

20.2 The Source of Steam Velocity

Let's assume I wish to generate 1 lb of saturated 400-psig steam in a boiler. I'll be starting with a pound of water at 60°F and atmospheric pressure. I'll do this in three steps and express the energy required for each step in Btu.

- Step One: Pump water up to 400 psig—requires about 1.2 Btu.
- Step Two: Heat water to its boiling point of about 440°F at 400 psig—requires about 380 Btu.
- Step Three: Vaporize the water to steam—requires about 820 Btu.

Any boiler plant operator trying to optimize steam generation efficiency will worry most about the firebox and convective section economizer, and not the boiler feed water pump. About two-thirds of the energy required to generate steam is devoted to latent heat and one-third to sensible heat. The energy needed to increase the pressure of the water is very small in comparison.

A hydroelectric power station generating electricity from a water-fall requires the flow from an entire river. A coal-fired power station generating electricity from steam requires a small water rate that could flow through a 12-inch pipe. The hydroelectric station is only using the potential energy of the water that has been converted to velocity. The thermal power station is using the heat content of the steam that has been converted to velocity.

20.2.1 How to Convert Heat into Motion

I have two ways of expanding steam. The bad way and the good way. In both ways I am not changing the energy content of the steam. I am not extracting any work from the steam; there is no friction and no heat is added or lost. This is called an adiabatic expansion.

Figure 20.1(a) illustrates the bad expansion. I expand 400 psi steam down to atmospheric pressure by going from a small pipe to a

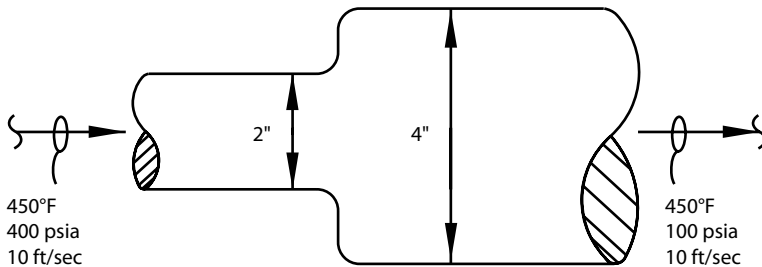
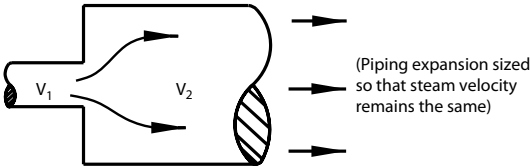


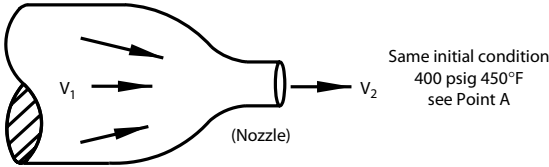
FIGURE 20.1(a) An isenthalpic expansion. The heat and speed of the steam remains constant.

Case 1 $V_1 = V_2$ Isoenthalpic expansion



- Velocity of steam status constant initial condition 400 psia 450°F (see point A)
- Follow line of constant enthalpy to atmospheric pressure line
- Then estimate final temp (°F) by reading between temp lines (see point "Case 1") approx. 330°F

Case 2 V_1 Less than V_2 isentropic expansion



- Follow lines of constant entropy to atmospheric pressure line (use point "Case 2")
- Read enthalpy at point A then subtract enthalpy at point "Case 2"
- Difference in enthalpy is 1210 – 970 = 240 Btu
- Final temp = 212°F

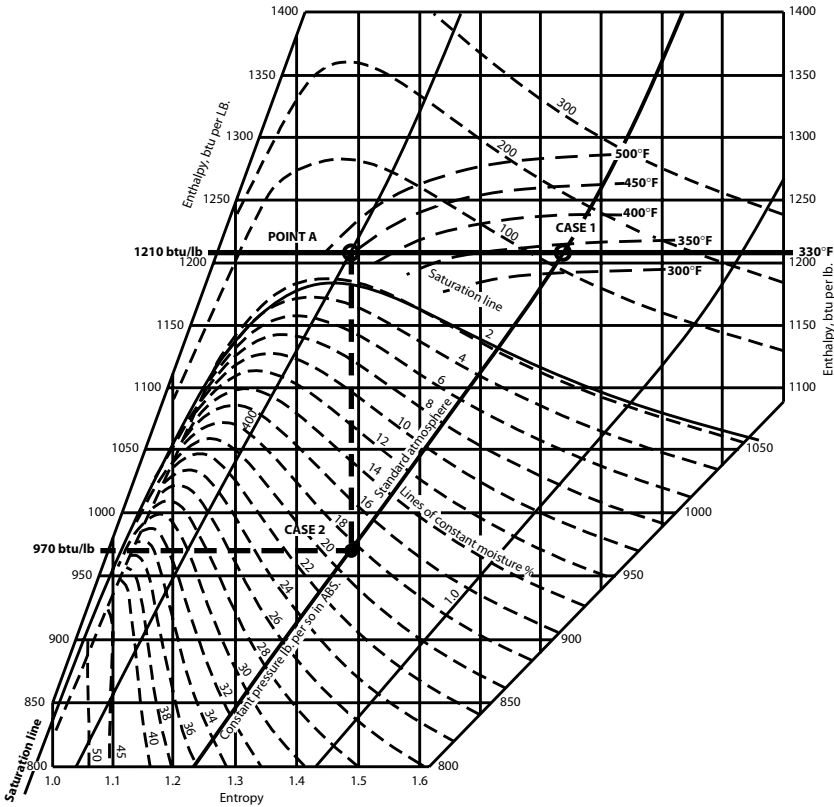


FIGURE 20.1(b) Using a Mollier diagram.

big pipe. That is, the cross-section area of the pipe increases to keep the steam velocity constant.

This sort of expansion is bad because it accomplishes nothing. The velocity of the steam has not increased at all. It's called an isenthalpic expansion because the heat content of the steam has not been reduced. None of the heat has been converted to velocity. And since it is the velocity—not the heat—of the steam that does work, nothing has been accomplished.

It's like taking a girl on a date to an expensive restaurant. At her front door, she says, "I had a wonderful time. Thanks!" What have you accomplished? Nothing!

20.2.2 Isoentropic Expansion

I like to call the isenthalpic expansion a parasitic expansion because it wastes the potential ability of the steam to do work. On the other hand, we have the good isoentropic expansion that maximizes the ability of the expanded steam to do work. Figure 20.1(b) - Case 2, also illustrates this sort of good expansion.

This type of expansion is good because it increases the velocity of the steam. Thus, the question is, what is the source of energy needed to:

- Increase the kinetic energy of the steam?
- Accelerate the steam?
- Increase the momentum (mass times velocity) of the steam?

This energy does not come from the steam pressure but from the temperature of the steam. We are converting the sensible heat content (thermal energy) of the steam to *kinetic energy*.

It's like taking a girl home from dinner and she says, "Why not come up for a cup of coffee?"

Comparing the two expansions in Fig. 20.1(b), the exhaust steam temperature has dropped from 330°F to 212°F, because the velocity has increased. The higher velocity steam can do more work than the slower steam. This single idea is the basis for the second law of thermodynamics, one of the fundamental principles that govern the dynamics of our universe.

Cooling of Steam on Expansion

When steam is reduced in pressure, while maintaining a constant velocity, its flowing temperature goes down. How can the steam temperature go down without:

- Increasing velocity?
- Extracting work from the steam?
- Heat losses to the environment?

The answer is that at a lower pressure the steam molecules move away from each other, which then lowers the frequency of molecules colliding with the temperature sensor. However, as the molecules move away from each other, the latent heat of condensation stored up by the molecules increases. In effect, just lowering the pressure without any change in velocity converts some of the sensible heat of the steam to latent heat of the steam, but without any effect on the heat content of the steam.

This is also true for light hydrocarbons (ethane, propane, butane), but not for hydrogen.

20.3 Converting Latent Heat to Velocity

Reducing steam pressure from 400 psi down to 100 psi in an isentropic manner, as shown in Fig. 20.2, will have two positive results:

- High velocity steam, at 100 ft/sec, that can do work
- One hundred psi steam that I can use to reboil distillation columns

But suppose I don't wish to use the 100-psi steam and would simply like to exhaust the steam to the atmosphere. Then, instead of the steam velocity being 100 ft/sec, it might exhaust from the discharge nozzle at 500 ft/sec. Where does all that extra kinetic energy come from?

- Some comes from a further reduction in the temperature of the steam. That is, the steam temperature will drop from the 440°F supply temperature down to 212°F [see Fig. 20.1(b)].
- However, most of the extra kinetic energy of the steam comes, not from the sensible heat, but from the latent heat of the steam. To be precise, 10 percent of the steam will condense to water.

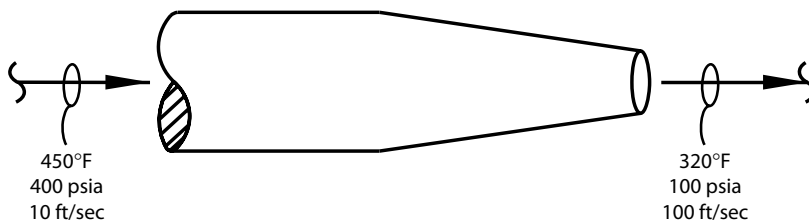


FIGURE 20.2 An isentropic expansion. Heat decreases and velocity increases.

We can extrapolate this concept further. Suppose I exhaust the steam to a vacuum of perhaps 25 inches of mercury (30 inches being a perfect vacuum). Now about 15 percent of the steam would be condensed and the steam (plus condensate) velocity might be 1000 ft/sec. I can extract much more work from each pound of steam. Of course, not only don't I have any useful steam left over, but I also have the expense of condensing the exhaust steam under vacuum conditions (see the Mollier diagram in Fig. 20.1).

It's like the girl you have taken out to dinner asking, "What do you like for breakfast? Just coffee, or toast and eggs?"

(Author's Note—The numerical values expressed above are for illustration only. They are not correct in an absolute sense. You may study these concepts and arrive at accurate conversion rates of steam enthalpy to kinetic energy by using the Mollier diagram in your steam tables. Just follow the vertical isentropic lines down the chart from the motive steam conditions down to the exhaust steam conditions. Note the lines of constant moisture as you drop below the saturated steam envelope line.)

20.4 Effect of Wet Steam

I suppose it may seem strange that creating more condensation of the motive steam could allow more useful work to be extracted from the steam. But it was this particular observation that led James Watt to vastly increase the efficiency of the steam engine almost 200 years ago. On the other hand, moisture in the motive steam supply itself is bad. As the steam is expanded to a lower pressure, the moisture in the supply steam will evaporate and cool off the steam, thus reducing the amount of sensible heat that could be converted to steam velocity.

20.4.1 Momentum in Steam Turbines

A useful way of summarizing the concepts I have just related is to think about *momentum*. That is:

$$\text{Momentum} = \text{Mass} \times \text{Velocity}$$

When we operate a steam turbine, we are trying to provide a certain amount of horsepower to drive our pump, or compressor, or generator. If I want to provide this power with less mass of steam, I need to increase the velocity of the steam. And this is done by extracting more sensible heat from the motive steam, and more importantly, more latent heat from the motive steam.

You can see quite clearly that it is not the pressure of the steam that is driving a turbine. Place a pressure gauge on the turbine case of a single-wheel machine. The pressure on the turbine case will be quite close to the exhaust steam pressure and certainly not the motive steam pressure.

20.5 Steam Ejector Temperature Profile

The entire concept of thermodynamics, the conversion of steam heat into steam velocity, can best be observed on a vacuum jet. Let's say I have 150-psig (10-BAR) saturated steam at 360°F flowing to the jet. As soon as the steam enters the ejector through the steam nozzle, the steam cools off to about 90°F. Don't take my word for it. Take your temperature gun and see for yourself. What happened to the missing 270°F of the steam (360°F–90°F)? It was converted to velocity!

As the steam slows down through the diffusor portion of the jet, you would expect the steam to partially heat, back up. And it does, perhaps to 200°F to 250°F at the discharge of the jet (i.e., at the condenser) inlet.

20.6 Roto-Flow Turbo Expander

I did not understand thermodynamics at university. It seemed to be an exotic subject with no relevance to my future as a refinery process engineer. It wasn't until I had spent many years working with steam turbines, surface condensers, steam distribution systems, and most especially vacuum ejectors that I realized that without a firm understanding of thermodynamics, I could not operate or troubleshoot such facilities with any degree of confidence.

For those of you who work in the liquefied natural gas industry, all of the above principles discussed relating to steam apply to light liquid hydrocarbons as well. The Roto-Flow Turbo Expander, which you use to partially liquefy the natural gas, is much the same as a steam turbine. That is, the motive gas is partially condensed to ethane and propane, as the temperature of the gas is converted to velocity, as the gas flows into the Roto-Flow turbine case. Much of the torque developed by the expander turbine is derived from the latent heat of condensation of the liquefied petroleum fractions.

20.7 The Meaning of Entropy

The second law of thermodynamics states that:

Change in entropy = Change in heat divided by temperature

Or

$$\Delta S = \Delta Q \div T \quad (20.1)$$

But what does this actually mean to ordinary people? Not people like Einstein or Edison or Marie Curie, but to people like you and me, who got a D in thermo?

Entropy is a measure of the amount of heat that's available at a particular temperature. Lots of entropy is really bad. Heat available at a low temperature, which means the heat has lots of entropy, is bad.

For example, let's say Paulie knows 100 girls in high school. Each girl likes Paulie enough to smile at him in the hall. This is like having a large amount of heat, but it's so diffuse that it's almost useless. Having lots of heat available at a low temperature means the heat is so spread out that we can't do much with it. Paulie's problem is that his girlfriends are suffering from excessive entropy.

On the other hand, Frank is involved with Cathy. All the other girls think that he's a nerd. But Cathy meets Frank after school so that they can study *The History of Art* in Frank's room. This is like having a small amount of heat, but it's so concentrated that it's really useful. Having a small amount of heat available at a really high temperature means the heat has only a little entropy, which is good.

Liz has just reviewed the preceding analogy and has accused me of "sexist stupidity." Thus, I'll offer a better example of entropy. Let's say I've got 1000 Btu worth of heat in a big pot. But the temperature of the pot is only 68°F. It's nice to have the heat, but at 68°F, I can't think of any way to use it to make money. The problem is my pot has too much entropy.

Alternatively, I've got a tiny pot with 100 Btu of heat. The temperature of this tiny pot is 2000°F. I can use this heat to:

- Make steam
- Generate electricity
- Distill wine into whiskey

All of which I can use to make money. My tiny hot pot has very little entropy.

Concentrating more heat in a smaller pot makes the stuff inside the pot hotter and reduces its entropy. And because concentrated heat with low entropy is easier to do things with than dilute, colder heat with high entropy, we say that:

"Increasing the entropy of heat reduces its ability to do work."

Ninety-plus percent of the energy of the universe is available at 3°K (−270°C). That's the background radiation that contributes to static on your radio. This is the heat left over from the "Big Bang," 13½ billion years ago. But it's useless because it's so cold that all that heat has too much entropy.

A small percentage of the universe's energy is stored inside stars (there are 4×10^{22}) in the form of gravity and hydrogen, which can fuse into helium, oxygen, carbon, iron, and people. That energy is available at tens of millions of degrees. It's really useful because it's so hot and thus has very little entropy.

A process that takes energy in the form of heat, and just lowers its temperature, must increase its entropy. This is bad.

A process that takes energy in the form of heat and converts that heat into momentum or kinetic energy does not increase its entropy,

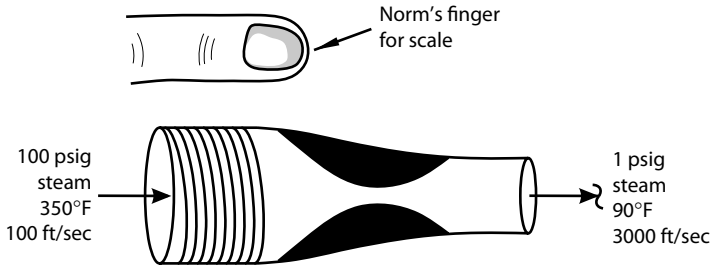


FIGURE 20.3 Steam nozzle used in a large power generator surface condenser first-stage steam ejector vacuum system.

because when I reduce the kinetic energy, I would get all the heat back at the original temperature, provided the entire process was:

- “Reversible”
- Adiabatic

Expanding high-pressure steam to a low pressure through a steam nozzle used in turbines and vacuum ejectors (see Fig. 20.3) is my favorite example of an isentropic process.

Almost all of the energy needed to accelerate the steam from 100 to 3000 ft/sec comes from cooling the steam from 350°F to 90°F. In an isentropic process, the heat content of the steam decreases, and its entropy is constant, provided that the process is fully reversible (Fig. 20.2).

In an isenthalpic process, the heat content of the steam remains constant and its entropy increases (Fig. 20.1).

The concept of entropy was not invented just to confuse young students by Herr Professor Clausius. It was intended to define the efficiency of steam engines. Any portion of a steam engine that permits steam to expand and cool, without an equivalent increase in the kinetic energy of the steam, will increase the entropy of the steam and also reduce the steam’s ability to do useful work, such as spinning a turbine’s wheel or pulling a vacuum.

When I used the word, *equivalent* in the last sentence, I meant converting an amount of heat energy to speed, kinetic energy, or momentum.

- Thermo = Heat
- Dynamics = Speed or Motion

For me, thermodynamics means “Heat in Motion.” You can buy this book (by Professor John Tyndall) in paperback from Amazon. I have an original sixth-edition copy (1880). The full title is *Heat, a Mode of Motion*.

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CHAPTER 21

Deaerators and Steam Systems

Generating Steam in Boilers and BFW Preparation

The one thing that almost all process and chemical plants have in common is steam. Steam is almost universally used to transfer energy over reasonably short distances (i.e., less than a mile), for the following reasons:

- Steam has a high latent heat of condensation. This permits flowing steam to transmit about eight times more energy than that transmitted by a pound of butane.
- Steam is nonexplosive, and hence reasonably safe to handle.
- Steam is generated from water, which, while not free, is often inexpensive to obtain.

Steam is used for several purposes in a modern process plant:

- To spin turbines
- To provide process heat, mainly to reboilers and preheaters
- As a stripping medium in fractionation towers
- To increase the velocity of heavier liquid streams, typically in furnace tubes
- As a *lift gas*, to lift heavier liquids or solid catalyst streams to a higher elevation
- As a purge medium, to prevent pressure taps and other instrument connections and relief valves from plugging

If energy for either motive or heating purposes must be transmitted over long distances (i.e., more than a few miles), it is best to use

electricity. The energy loss (due to both friction and ambient-heat radiation) is orders of magnitude greater for steam as compared to electrical power. Most electricity is produced from steam by use of a steam-powered generator. Typically, it takes 3 Btu of steam to generate 1 Btu of electricity. That is why it is best to use the steam directly, at least for heating purposes, if the distance between the boiler house and the consumer is small.

21.1 Boiler Feedwater

21.1.1 Sources of Boiler Feedwater

Steam is generated from *boiler feedwater*. But water used for steam generation must first be purified. For example, raw water drawn from the Mississippi River has three objectionable contaminants:

- Sand and silt
- Dissolved solids (typically carbonates)
- Oxygen

The particulate matter (sand and silt) is removed by flowing upward through a gravel-and-sand filter bed. The coarser gravel is in the lower part of the bed. The filter beds are back-flushed periodically with clear water to clean off the accumulated mud.

The dissolved solids, such as calcium carbonates, are removed by hot-lime softening or demineralization.¹ Demineralized water (also called *deionized water*) typically has essentially all anions and cations removed by ion-exchange resin. Demineralized water is preferable to hot-lime-softened water as boiler feedwater for several reasons.

For one thing, steam produced from hot-lime-softened water will have some amount of silicates. These silicates tend to deposit on the rotor blades of turbines, which use the motive steam as a source of energy. The silicate fouling of the turbine blades reduces the turbine's efficiency. But, more importantly, from an operator's point of view, the silicate deposits eventually break off of the blades. This unbalances the rotor. An unbalanced rotor is the fundamental cause of vibration. Vibrations lead to damage of the shaft bearings and seals. Eventually, vibrations will destroy the turbine's internal components.

Also, the hot-lime-softened water has variable amounts of carbonate contamination. When boiler feedwater is converted to steam, the carbonate deposits will break down into carbon dioxide and hardness deposits.

The hardness deposits coat the inside of the boiler's tubes, interfere with heat transfer, and overheat the tubes. The carbon dioxide, which is also generated from the dissolved solids, creates more serious corrosion problems in downstream heat exchangers. When the

steam condenses, the carbon dioxide may remain trapped in the reboiler or preheater as a non-condensable gas. Actually, there is no such thing as a non-condensable gas. Even CO_2 is somewhat soluble in water. As the CO_2 dissolves in the condensed steam, it forms *carbonic acid*, a relatively weak acid (pH typically between 5 and 6). Strong acids will have pH values of 1 to 2. Pure water has a pH of 7. Carbonic acid is corrosive to carbon steel heat-exchanger tubes.

Oxygen is a particularly reactive element. Actually, oxygen is a very potent gas and reacts aggressively with exposed metal surfaces to form oxides. Of all the corrosive substances encountered in a process plant, few exceed oxygen in reactivity with steel pipes. If a considerable amount of oxygen is left in boiler feedwater, interior corrosion of the boiler tubes will be rapid. The bulk of the oxygen, or dissolved air, is stripped out of the boiler feedwater in a *deaerator*.

In properly operated process plants, most of the boiler feedwater should be recovered *condensate*; that is, when steam condenses to water, the condensate produced should be recovered and returned to the boiler house. Often, the recovered condensate will first pass through a deoiling clay bed to remove any hydrocarbon contaminants. To calculate the percent of condensate recovered, we need to know two numbers:

- Total steam production (SP)
- Total raw water flow (RW)

The percent condensate recovery (CR) can then be calculated:

$$\text{CR} = 100\% - 100 \times \frac{\text{RW}}{\text{SP}}$$

Plants with CR >70 percent are usually doing a good job of recovering condensate. CR values of <30 percent represent poor performance. Poor condensate recovery is expensive for two reasons. One reason is the energy and chemicals needed to prepare the boiler feedwater. But in recent years, there is a second, more expensive cost associated with poor condensate recovery: effluent wastewater treatment.

I well remember one Louisiana refinery in which boilers and heat exchangers suffered from constant steam-side corrosion and fouling. Investigation showed that the intake water-treatment plant operators would occasionally partially open the raw-water bypass around the entire treatment plant. They had a good reason, however, which they explained to me. "Norm, y'all have to admit, some kind of water is better than no water at all."

21.1.2 Deaerators

Figure 21.1 is a sketch of a typical boiler feedwater deaerator. The lower portion of the deaerator acts as a surge vessel for the boiler charge pump. The working part of the deaerator is the smaller vertical

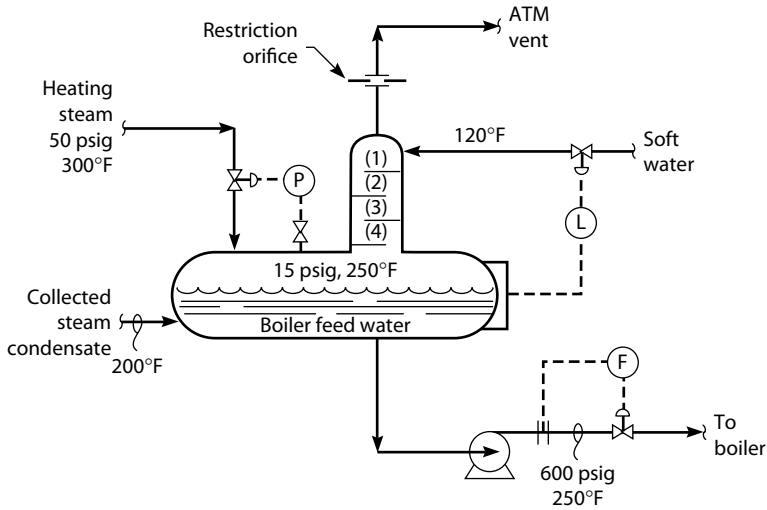


FIGURE 21.1 Deaerator is intended to remove O_2 from boiler feedwater.

section on top of this surge vessel. The vertical section is essentially a small, four-tray, steam stripper tower (see Chap. 15). Sometimes these trays are called *mixing shelves*, but their function is the same as any distillation tower tray: to bring the vapor (steam) into intimate contact with the liquid (soft water). The objective of the little tower is to drive off the light component (air, dissolved in the soft water) with the heavier component (steam).

The deaerator shown in Fig. 21.1 is operating at 250°F. This particular temperature corresponds to the boiling point of water at 15 psig. We can say, then, that the temperature of the deaerator controls its pressure. But it is equally correct to say that the pressure of the deaerator sets its temperature.

The 120°F soft water is heated to 250°F by the 50-psig heating steam. In most deaerators, the vast majority of the heating steam is condensed inside the deaerator. The heat of condensation of the steam is used to heat the soft water.

Usually, only a small amount of the heating steam—specifically, *stripping steam*—is vented through the restriction orifice located in the atmospheric vent. It is this small stripping steam flow that drives the air out of the soft water.

Note that the collected steam condensate does not pass through the little stripper section. As this stream is just condensed steam, it should be free of air. Hence, it does not need to be stripped.

21.1.2.1 A Common Deaerator Malfunction

Big Jim was once a high school teacher in New Orleans. I do not quite understand the connection, but many high school teachers become

really good process plant operators. Perhaps they find the work less hazardous. Big Jim became an operator at the utility complex for a major New Orleans-area sugar refinery. Here is a rather neat problem that Jim and I solved together.

The problem was cavitation of the high-pressure boiler feedwater charge pump, shown in Fig. 21.1. Such pumps are often multi-stage. This means that there are three or four pump impellers on a relatively long shaft. The longer the pump shaft, the greater the damage caused by vibration to the pump support bearings and mechanical seal. Cavitation is certainly the leading cause of vibration in centrifugal pump operation (see Chap. 36). The cause of cavitation in this case was loss of liquid level in the deaerator surge or water storage vessel.

The symptoms of the problem Jim described to me were as follows:

1. An operator working at the main boiler would open the boiler's blowdown valve. Some lime-softened water, about 10 percent of the boiler's total feedwater must be drained to maintain the total dissolved-solids (TDS) content of the boiler's circulating water below a safe maximum.
2. This would draw water out of the deaerator vessel.
3. The level in the deaerator would fall, and the flow of 120°F soft water into the deaerator would increase.
4. Suddenly, the flow of softened water would increase exponentially. The level-control valve would rapidly open. Yet, the level in the deaerator would slowly continue to fall until the pump lost suction and cavitated.

Jim and I then made the following field observations:

1. When the flow of softened water exceeded 80,000 lb/h, the heating steam pressure-control valve, shown in Fig. 21.1, would open to 100 percent.
2. A further increase in softened water flow drove the deaerator pressure down. This happened because the deaerator temperature also dropped.
3. Suddenly, the atmospheric vent pipe, which had been venting steam, started to blow out rather large quantities of water with the steam.
4. This removed some of the water from the boiler feed surge drum portion of the deaerator.
5. As the deaerator water level fell, the 120°F softened-water flow increased.
6. The escalating flow of colder softened water cooled down the deaerator.

7. Since the water in the deaerator must always be at its boiling point, the reduction in the deaerator temperature resulted in a reduction in the deaerator pressure.
8. The reduced deaerator pressure coincided with an increase in water jetting out of the atmospheric vent.

What was happening?

Well, we have already discussed this phenomenon in Chap. 3 in the section on tray flooding. Water carry-over from the top of tray 1, as shown in Fig. 21.1, is simply a clear signal that our little four-tray distillation tower is flooding. But what is causing the trays to flood?

Trays usually flood because of excessive vapor flow. The vapor flowing to tray 4 is essentially the heating steam flow. Certainly, most of this steam condenses on the trays, as the heating steam comes into contact with the 120°F softened water. It takes about 1 lb of 50-psig steam to heat 7 lb of water from 120 to 250°F. The uncondensed steam is then vented from the top of our tower through the atmospheric vent. But the full steam flow, from the pressure-control valve, does flow up through tray 4. Obviously, when this pressure-control valve is 100 percent open, the weight of vapor flow to tray 4 is at its maximum.

But, dear reader, do not forget that when the steam inlet pressure-control valve is 100 percent open, any further increase of cool softened-water flow will suppress the deaerator's pressure. When the pressure of a vapor goes down, its volume goes up:

$$\text{Volume} \sim \frac{1}{\text{Pressure}}$$

And when the volume of a vapor flowing through a tray increases, so does its velocity. Any increase in vapor velocity through a tray results in higher tray pressure drop. And what is it that causes trays to flood? Why, it is high tray deck pressure drop.

To solve this problem, Jim recommended to his boss that the softened water be preheated by using it on one side of a heat exchanger to cool off a waste foul liquor stream that was flowing to the effluent treatment pond. This heated the softened water from 120 to 160°F. The 50-psig steam demand was reduced by 25 percent. This reduced steam flow prevented the trays from flooding.

Note that it is okay to heat softened or demineralized water to 160°F. You should not, however, do this with raw water. Calcium carbonate salts will plate out and foul the heat-exchanger tubes.

I rather like this problem. It illustrates how knowledge of one type of process equipment (a distillation tower) can help us understand problems in a piece of equipment that is superficially quite a bit different.

21.2 Boilers

A rather typical boiler is shown in Fig. 21.2. This is the sort of boiler one might rent to generate 50,000 lb/h of 150-psig steam. Incidentally, rented process equipment does not have to be returned to the owner if the plant files for bankruptcy. But that is another story.

The boiler shown in Fig. 21.2 relies on natural circulation. The density difference between the water plus steam in the downflow pipes and the riser pipes causes the water to circulate through the pipes. A typical circulation rate results in 10 to 15 percent of the water flow being vaporized into steam. The produced steam flows from the steam drum into the steam header. The water level in the steam drum is maintained by the makeup boiler feedwater valve.

A certain amount of water is drained from the mud drum. The amount of this blowdown is determined by measuring the TDS in the steam drum. Typically, a target of several thousand parts per million (ppm) of TDS is specified. If the steam is generated at a low pressure (150 psig) for use in reboilers, rather high TDS may be acceptable. If the steam is generated at a high pressure (1400 psig) for use in high-speed turbines, a much lower TDS target would be set.

It is really the quality of the treated boiler feedwater that sets the blowdown rate. Deionized or demineralized water might require a 1 to 2 percent blowdown rate. Hot-lime-softened water might require a 10 to 20 percent blowdown rate.

The amount of hardness deposits and silicates in steam is a function of entrainment of water, as well as the TDS of the boiler

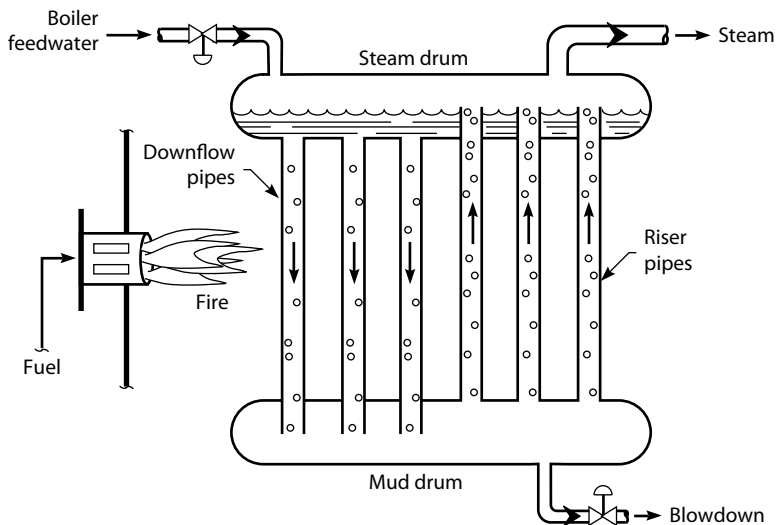


FIGURE 21.2 Natural-circulation boiler.

feedwater. A well-designed boiler, then, often is equipped with a mesh-type demister pad to remove entrainment from the produced steam.

The water level in the steam drum in Fig. 21.2 controls the boiler feedwater (BFW) makeup rate. However, note that the steam drum is full of boiling water. How can we measure the real level of this two-phase mixture in the steam drum?

After all, the external level indication chamber (level-trol) holds just water. When we look at the gauge glass, it has water, not steam. Hence, the indicated water level outside the drum is less than the boiling water level inside the drum. This discrepancy causes water carry-over with the steam due to high boiling water levels.

If we are superheating the steam, then the silicates and other salts in the entrained water will deposit inside the superheater tubes. What I've just described is the leading cause of superheat tube failure. Most of the trouble can be avoided if you understand that the actual level inside the steam drum is higher than the external water indication shows.

I've explained this matter in greater detail in the section on waste-heat boilers (Sec. 21.2.2).

21.2.1 Superheaters and Economizers

Figure 21.2 is not a complete picture of a larger industrial-type boiler. Mainly, it does not show the superheat and economizer tubes. Figure 21.3 gives a better idea of the relative arrangement of the steam-generating tubes, superheat tubes, and economizer tubes.

Let's assume that we are generating 100,000 lb/h of 150-psig steam. In the economizer section, the effluent from the deaerator, at 250°F, would be heated to 350°F. As the specific heat of water is 1.0 Btu/[(lb)(°F)], we would need 100 Btu/lb of water. However, to produce 100,000 lb of steam, we might need a 10 percent blowdown rate to control TDS. This means 110,000 lb/h of boiler feedwater is needed. Therefore, the economizer heat duty would be

$$110,000 \text{ lb/h} \times 100 \text{ Btu/lb} = 11,000,000 \text{ Btu/h}$$

The latent heat of vaporization of 150-psig steam is roughly 880 Btu/lb. Therefore, the steam-generation section heat duty would be

$$100,000 \text{ lb/h} \times 880 \text{ Btu/lb} = 88,000,000 \text{ Btu/h}$$

The specific heat of steam is 0.5 Btu/[(lb)(°F)]. As shown in Fig. 21.3, the saturated steam will be superheated from 360 to 680°F. Or, we may say that the steam superheat will be 320°F. We would then need 160 Btu/lb. Hence, the superheater section heat duty would be

$$100,000 \text{ lb/h} \times 160,000 \text{ Btu/lb} = 16,000,000 \text{ Btu/h}$$

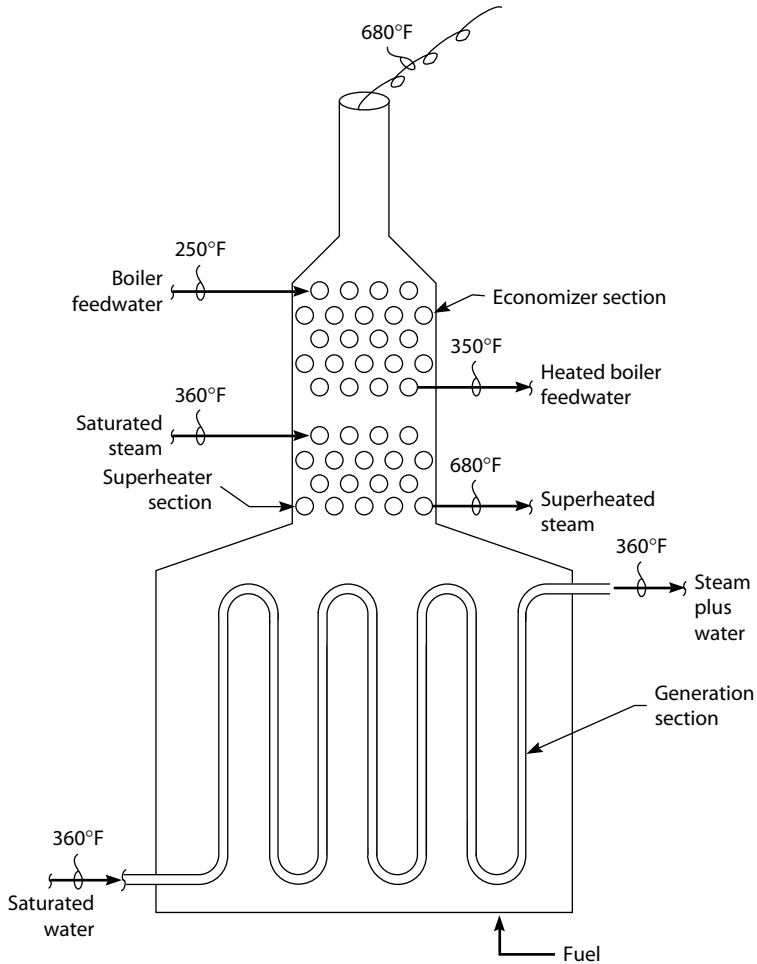


FIGURE 21.3 Simplified drawing of a boiler showing economizer and superheat sections.

The total boiler heat duty would be the sum of the economizer, generation, and superheat sections, or 115,000,000 Btu/h.

I have noted that the stack flue-gas temperature of the boiler is 680°F. Let's assume that we have around 3 percent oxygen in the flue gas (or about 15 percent excess air). We can then apply the following rule of thumb: *For every 300°F that the flue-gas temperature exceeds the ambient air temperature, the boiler efficiency drops by 10 percent.*

If our ambient air temperature is 80°F, then our boiler efficiency is 80 percent. The fuel consumed in our boiler is then

$$\frac{115 \times 10^6 \text{ Btu/h}}{0.80} = 144,000,000 \text{ Btu/h}$$

21.2.2 Waste-Heat Boilers

Probably only 20 percent of process plant operators or engineers ever work on the sort of boilers shown in Figs. 21.2 and 21.3. However, 90 percent of process plant personnel will, at one time or another, in all likelihood operate a *waste-heat boiler*. In many petroleum refineries and chemical plants, most of the plant's steam is generated in such waste-heat boilers. One of the most common such boilers is the kettle boiler, shown in Fig. 21.4.

The hot oil to the kettle boiler is a circulating pumparound stream from a fluid catalytic cracker fractionator, slurry-oil circuit. There is a fundamental difference between this sort of boiler and the utility plant boilers discussed previously. In the kettle boiler, the heating medium is inside, rather than outside, the tubes. To obtain the full capacity of the kettle boiler, the uppermost tubes should be submerged in boiler feedwater. This requirement creates a few problems.

First, note the water level in the level indicator shown in Fig. 21.4. It looks quite a bit lower than the water level inside the kettle. Why? Well, because the water inside the kettle is boiling water. The water density has been reduced by the bubbles of steam. I have discussed this problem in great detail in Chap. 10 in the section on level control. Basically, the difficulty is that the water in the level indicator has a specific gravity of about 1.0. The boiling water in the kettle has a specific gravity of perhaps 0.5. If we have 4 ft of boiling water in the kettle, then, by pressure balance, we will have 2 ft of cool water in the level indicator.

To compensate for this problem, the top tube of the tube bundle should be kept well below the top of the kettle. This also will help retard entrainment of boiler feedwater in the steam.

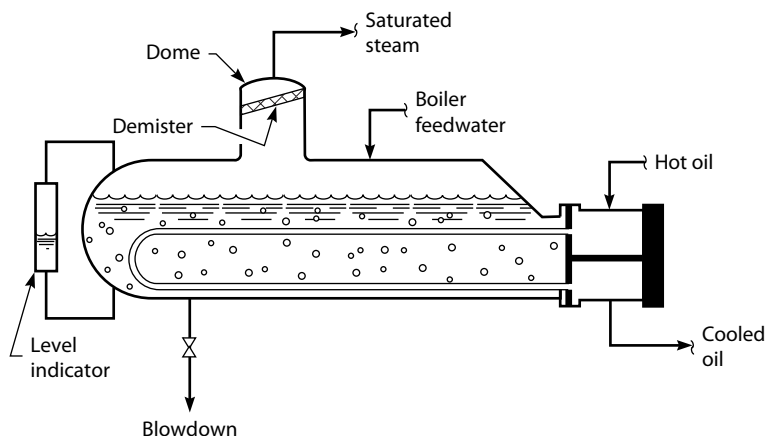


FIGURE 21.4 Kettle waste-heat boiler.

To further improve the steam quality, the experienced design engineer places a dome, as shown in Fig. 21.4, on top of the kettle. The dome will have a sloped demister mesh pad. A *demister* acts to coalesce small droplets of water into larger droplets. The larger droplets easily settle out of the flowing steam. The demister is sloped to promote liquid drainage from the surface of the pad.

Of course, if the steam from the kettle will be used to reboil some nearby column, entrainment is not a major problem. If the steam is going into superheater tubes, however, serious entrainment will lead to solid deposits inside the superheater tubes, and this may cause these tubes to overheat and fail. Fouling of turbine blades would also result.

21.2.3 Identifying Waste-Heat Boiler Tube Leaks

Fig. 21.5 shows two waste-heat boilers that are generating superheated, 150-psig steam from a fractionator hot oil pumparound. After years of trouble-free operations, water was observed in the gas-oil pumparound return stream. The symptoms of the water infiltration into the fractionator were an increase in the fractionator pressure and an increase in the water production from the fractionator reflux drum's draw-off boot.

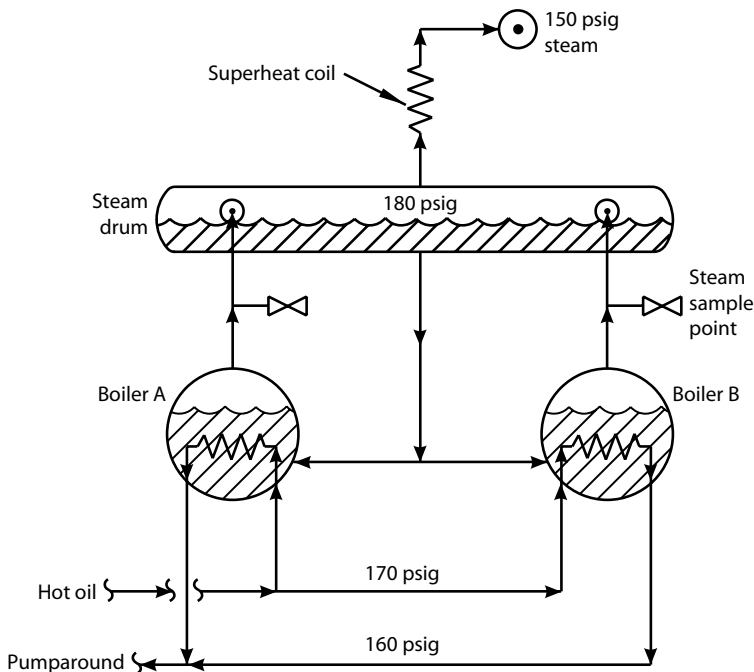


FIGURE 21.5 Finding which waste-heat boiler is leaking may be confusing.

The operators planned to bypass boilers A and B in turn, as shown in Fig. 21.5, to determine which boiler had a tube leak. Boiler A was arbitrarily selected. The hot gas-oil pumparound from the fractionator was closed. With the reduction in the gas-oil pumparound heat removal, the generation of 150-psig steam dropped in half, and the water intrusion into the fractionator stopped. The operators concluded that with waste-heat boiler A out of service, and the water leak into the fractionator apparently eliminated, the leaking boiler was A.

However, when boiler A was hydrotested by the maintenance department, no leak was found. So boiler A was returned to service. But, as the operators restored the flow of gas oil to boiler A, the indications of water intrusion into the fractionator returned. The operators, quite angry, again removed boiler A out of service and insisted that the maintenance department this time find and actually fix the tube leak in boiler A.

This scenario was repeated three times, until the plant manager intervened and instructed the engineering department to purchase a new \$250,000 tube bundle for boiler A. The refinery had lost \$100,000 per day in production for about 40 days, up to the day that the new tube bundle arrived. The new bundle was quickly installed. As the gas-oil pumparound flow was restreamed, the flow of water into the fractionator was once again initiated.

Can you see the problem? It's a failure in applied logic. Boiler A was never leaking. All along, it was boiler B that leaked. Here's what had happened:

- When boiler A was removed from service, the steam flow from the steam drum dropped by 50 percent.
- The ΔP through the superheat coil, shown in Fig. 21.5, dropped from 30 psi to:
 - $(\frac{1}{2})^2 \cdot (30) = 7\frac{1}{2}$ psi (i.e., ΔP varies with the flow squared).
- The pressure in the steam drum then fell from 180 psig to:
 - $150 + 7\frac{1}{2} = 157\frac{1}{2}$ psig (i.e., the pressure of the 150 psig plus the superheat coil ΔP).
- The pressure in the steam drum fell below the 160 psig pressure of the gas-oil pumparound return.
- The leak of water into the gas oil stopped. The leak reversed and became a small leak of gas oil into the steam drum.
- Whenever boiler A was returned to service, the extra steam flow repressurized the steam drum, above the pressure of the gas oil, and thus restored the water leak into the gas oil.

Liz and I did find that the steam was contaminated with gas oil whenever the waste-heat boiler B was in service with boiler A offline. We put boiler A back online again, and the water reappeared in the fractionator, as it had on the four prior occasions. However, we

now had boiler B isolated, but left boiler A online, with no resulting oil contamination of the 150-psig steam and with no water contamination of the gas-oil pumparound. A tube leak was found and fixed in boiler B, and then boiler B was also returned to service and the full unit charge rate was restored.

Liz and I solved this problem in just four hours, while my client failed to solve the problem in 40 days. One key observation we made was the oil contamination of the steam at the steam sample point shown in Fig. 21.5, when only the boiler B was in operation. The second observation we made was that the steam drum pressure increased above the gas-oil pumparound pressure, when both boilers were in service together.

This story illustrates the importance of making direct field observations. The relevant pressure gauges and samples were not available until Liz and I showed up onsite and asked that they be installed. It also illustrates an even more important principle of taking an analytical approach to problem solving. That is, all possible contributions to a problem need to be considered. Then field observations should be made to confirm or refute the existence of each problem.

21.3 Convective Section Waste-Heat Steam Generation

Figure 21.6 shows another common form of steam generation from waste heat. Boiler feedwater is circulated, via a pump, through the convective tubes of a fired heater. Most of the heat released by the fuel

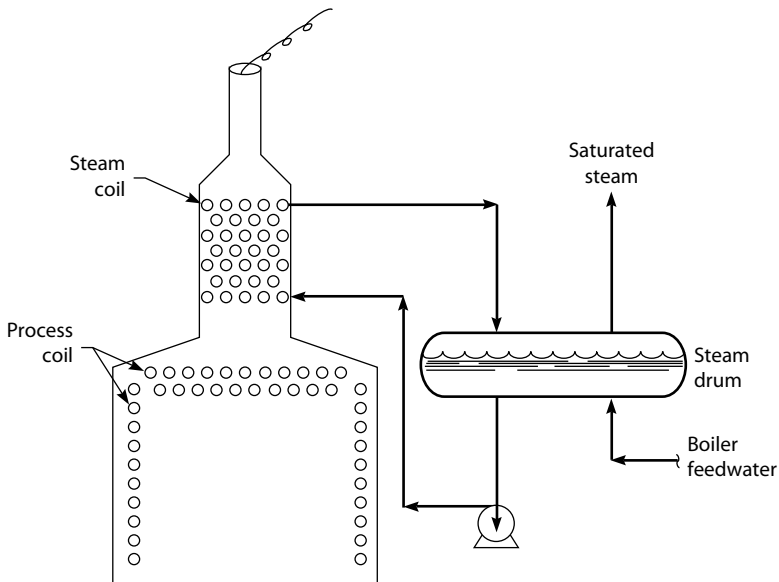


FIGURE 21.6 Convective section steam generation from waste heat.

(50 to 70 percent) is absorbed in the process coils. Perhaps 10 to 25 percent of the waste heat in the flue gas is absorbed in the steam-generation coils. It is good design and operating practice to keep the percent of water vaporized in the steam-generating coils to well less than 50 percent. Higher rates of vaporization promote the laydown of salts inside the convective tubes. Low water circulation rates will lead to tube leaks.

It is probably best to consume steam generated in small waste-heat boilers locally in steam reboilers and preheaters. Turbine drivers are best powered by boiler-house steam, where steam quality is more closely monitored and controlled.

References

1. *Betz Handbook of Industrial Water Conditioning*, 8th ed., 1980, Betz Laboratories, Inc., Trevose, PA 19047 (USA).
2. N.P. Lieberman, *Troubleshooting Process Plant Control* (Wiley, 2009).

CHAPTER 22

Steam Generation

An important feature of steam generation in a process plant is steam condensate recovery. This is usually a function of proper design of the steam-condensate controls on individual steam heaters and reboilers, as well as proper design on the condensate recovery system, as discussed in previous chapters. I've also dealt with this subject in great detail in my book, *Process Equipment Malfunctions* (McGraw-Hill, 2010).

In my experience, a reasonably good condensate recovery rate in a large, complex refinery is 70 percent. A substantial portion of the steam is used for process steam stripping, purging vessels, instrument connections, flare dispersal, and other nonrecoverable uses. Thus, the 70 percent rate is really rather good.

On the other hand, 30 percent condensate recovery as a percentage of the steam generation rate is rather bad and represents sloppy operations and bad condensate recovery design practices. Very approximately, 10 percent of the cost of steam generation could be saved by recovering the condensate. The condensate contains few silicates, thus no blowdown (explained below) is needed, no chemical treatment or deaeration is required, and no energy that would otherwise be used to preheat the boiler feedwater will be wasted.

22.1 Boiler Blowdown Rate

Although boiler feedwater (BFW) has been treated to remove minerals, residual contaminants (carbonates, silicates) are still present. These minerals, especially the silicates, are bad, as to some extent they are going to be entrained into the generated steam. The biggest problems with entrained silicates (TDS—Total Dissolved Solids) are:

- Drying out on the steam superheat coils, which then fail due to localized overheating, which results in superheat tube failure.
- Depositing on turbine blades. The deposits eventually get so thick that they break off and unbalance the rotor. Resulting vibrations then cause the turbine to trip-off, and stress corrosion of the blades causing sudden failure.
- Fouling of vacuum system ejector steam nozzles.

The TDS content of BFW should be minimized by upstream treating. In former days, hot lime water softening was employed. In most modern process units, ion exchange resin is used to demineralize BFW. “Blowdown” is the water that is drained out of the boiler to control the accumulations of TDS or silicates in the boiler. My old design value for this flow was 10 percent of the makeup BFW flow. Two types of blowdown from a boiler are used: continuous blowdown from the steam drum, and intermittent blowdown from the mud drum; both will reduce silica. Note that with the continuous blowdown, heat recovery equipment may be economically justified.

Continuous Blowdown—This blowdown water is best not drained from the very bottom of the steam drum. The concentration of silicates is 10 times higher at the steam-water interface than it is at the bottom of the steam drum. I’ve made the mistake in designing waste-heat kettle-type boilers of locating the drain on the bottom of the kettle, which is wrong for the same reason. The correct drain location is somewhat below the minimum water level in the boiler steam drum, or in the kettle-type boiler. TDS in the blowdown is based on what the steam will be used for. A typical value is 1500 ppm. If steam from a kettle waste-heat boiler will be used saturated, for heating, reboiling, or stripping, then the TDS content of the steam is not particularly critical. If steam is used to drive turbines or will be superheated, then the TDS content of the steam is quite important. A good target for silica is 0.02 mg/l silica in steam used for turbines or for a boiler operating from 300-psig to 600-psig 0.4 total caustic alkalinity as mg/l of SiO_2 in BFW.

Intermittent Blowdown—This blowdown from the mud drum of a boiler is usually once a shift via manually operated valve. If this is automated it is important to have the valve full open for short periods to keep the valve clear and unblocked.

22.2 Types of Steam-Generating Equipment

Figures 22.1, 22.2, and 22.3 illustrate the three types of steam-generating equipment that are used in process plants. Most process units do not have the boiler shown in Fig. 22.1. Water flows via thermosyphon circulation (i.e., the density difference between the steam drum and the mud drum). This is the kind of design used for a large boiler of the type found in the Utilities Department of a refinery or power plant.

Figure 22.2 is a convective section heat-recovery forced-circulation boiler. Many process-fired heaters have this feature. Hot flue gas from the radiant section is exchanged against BFW to make steam. A good water circulation rate is 10 times higher than the steam generation rate. Vaporizing 20 percent of the circulating water is still okay. Vaporization rates above this level will eventually cause convective section tube failures, as will even very short pump outages.

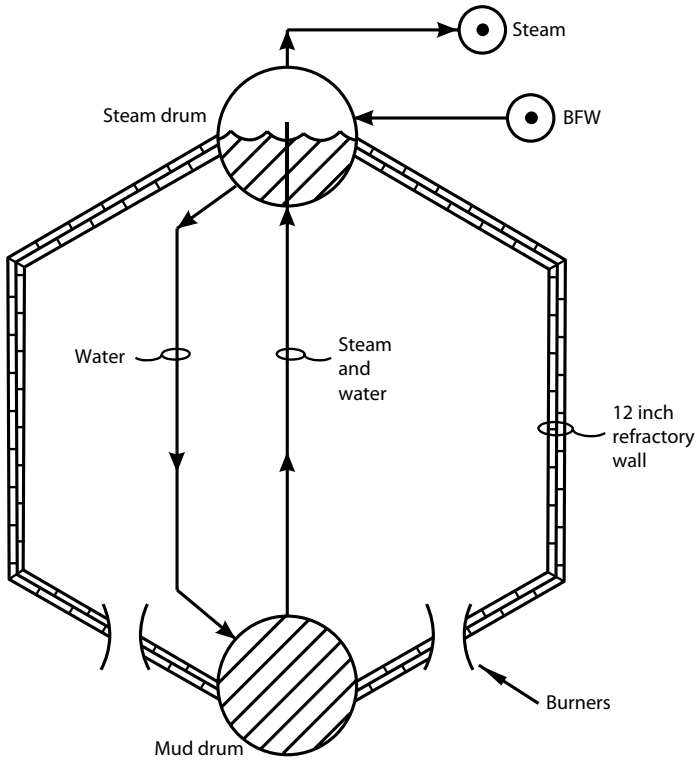


FIGURE 22.1 Fired boiler water circulates by thermosyphon circulation through tubes.

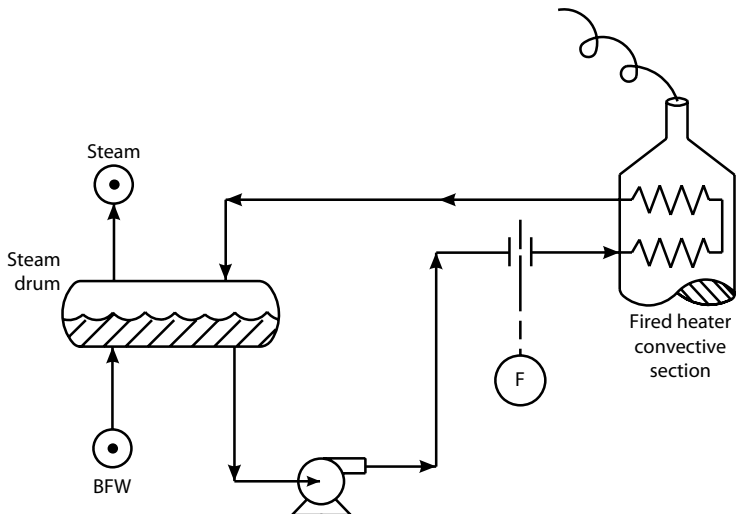


FIGURE 22.2 Heat recovery from flue gas-water circulates by forced circulation through convective tubes.

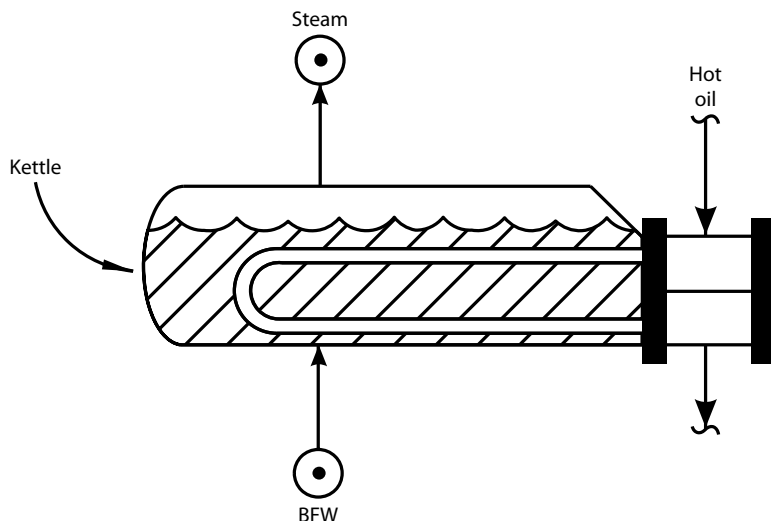


FIGURE 22.3 Waste heat recovery from hot oil—no water circulation. Tubes covered with BFW.

Figure 22.3 is the very common kettle-type waste-heat recovery boiler. There is no water circulation. The tubes must be covered by water at all times. Allowing the upper rows of tubes to be uncovered will lead to uneven thermal expansion of the tube bundle and damage due to distortion of the uncovered tubes. If the BFW level is too high, this will cause water and silicate carry-over into the evolved steam. This sort of carry-over leads to more steam turbine vibration failures than any other cause. It's one of the major problems in the process industry.

22.2.1 Controlling Steam Drum Levels

Ordinary level control does not work in steam drums or kettle boilers. The problem is the lower density of the boiling water as compared to the greater density of the non-boiling hot water in the external level sensing chamber. I've addressed this problem in my book, *Trouble-shooting Process Plant Control* (Wiley, 2009).

The boiling water will be at a 10 to 50 percent higher level inside the steam drum than the water in the level-sensing chamber or gauge glass. There are two ways of dealing with this problem:

1. The conventional method is to get a sample of steam condensate and measure it for TDS. If the blowdown water has a TDS of 1000 ppm and the sample has a TDS of 30 ppm, then the moisture content of the steam is 3 weight percent. The assumption is that the TDS is totally due to entrainment.

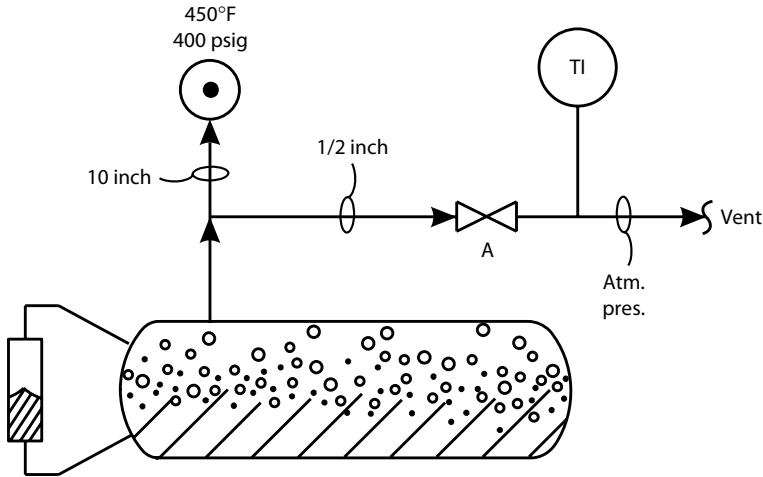


FIGURE 22.4 Steam drum level indication malfunction. Water level in drum higher than external level indication.

2. My method (which hopefully you will agree is better) is explained in Fig. 22.4:
 - Assume initially the 400-psig steam is dry but saturated at 450°F.
 - Depressuring this steam through valve A will reduce the steam temperature to approximately 360°F.
 - As the boiling water level rises and entrains into the 10-inch steam line, the temperature of the vented steam at T1 drops to 300°F.
 - Latent heat of water = 1000 Btu/lb.
 - Specific heat of steam = 0.5 Btu/lb/°F.
 - Therefore the water content of the 400-psig steam:

$$(360^{\circ}\text{F} - 300^{\circ}\text{F}) \times 0.5 \div 1000 = 3 \text{ wt } \%$$

Three weight percent water in steam is a lot, and would be considered to be poor-quality steam.

22.2.2 Superheating Steam

I've had some very bad experiences with superheating steam furnace convective tube failures. The cause of tube failure was overheating of the tubes due to hardness deposits precipitated from poor-quality steam. The convective section tubes will typically be carbon steel, which fails at about 1000°F.

The one case I have in mind occurred at a large refinery in northern Spain. I had supplied the retrofit design for their crude

unit distillation tower fractionation trays. The start-up went well and my new tray design worked fine—for two days. Then the furnace steam superheat coil failed and the unit was shut down. During the second start-up, the operators inadvertently filled the tower to the top with crude oil and damaged all the trays. The tower would no longer fractionate. I was blamed for a poor tray design.

The worst part about the whole incident is not that I was blamed for my client's operating errors. It was that the steam was being superheated for no purpose. All of the steam was being used for hydrocarbon product stripping, where steam superheat is not important.

For example:

- Product stripped is 450°F gas-oil
- Steam rate = 2 wt % on feed
- Superheated steam = 650°F
- Specific heat of steam = 0.5 Btu/lb/°F
- Specific heat of gas-oil = 0.65 Btu/lb/°F

Therefore, the temperature increase of the gas-oil due to the superheated steam is:

$$(650^{\circ}\text{F} - 450^{\circ}\text{F}) \times 2\% \times (0.5/0.65) = 3^{\circ}\text{F}$$

The 3°F is hardly worth the investment in the superheat coil.

Superheating steam for reboilers and steam heaters is mostly a waste, as it's a standard industry practice to inject condensate into the steam to bring the steam back to its saturation temperature. Superheated steam does not transfer heat as well as saturated steam. Hence, the de-superheating stations. As there is typically another convective section (usually BFW preheat) above the steam superheat tubes, no net energy savings result from steam superheat.

The maximum allowable superheat coil outlet temperature is 700°F to protect the carbon steel tubes from failure. At low steam generation rates, I have seen furnace firing rates and thus heater charge rates reduced to keep from overheating the steam above 700°F. If the steam is not used eventually to drive turbines, or as a reactant in a catalytic process (steam-methane reformer for H_2 or NH_3 production), then the heater capacity is being limited for no logical reason.

22.3 Boiler Feed Water Preparation

22.3.1 Condensate Polishing

The recovered steam condensate may often be contaminated with oil. As the condensate will flow into the lower portion of the deaerator, this possible oil contamination must be removed to protect the boiler

tubes from the formation of non-wettable films on the heat transfer surfaces which interfere with the detachment of bubbles so that the boiler tube surface is then not sufficiently cooled by boiler water thus causing severe corrosion and tube failure.

Low-level oil contamination (5 mg/l) can be removed by passing the condensate through clay, or diatomaceous earth, either as a bed supported by anthracite, or diatomaceous earth precoat filters supported by filter paper, porous stone, or wire cloth etc. In addition, a small amount of earth is fed continuously with incoming condensate at the filter inlet. The filter cake is later removed by back washing and rinsing.

Condensate with heavy oil contamination can first be treated by oil skimming or settling, or otherwise must be simply sent to the effluent treating facility.

Note that although it is beneficial to keep condensate hot in the condensate recovery system so as to reduce the solubility of air and oxygen (dissolved oxygen being a major cause of corrosion in condensate piping systems), oil is easier to remove from condensate when it is cool. Thus cold boiler feedwater is often used to cool the condensate stream, or air cooling is also sometimes used. Oil settling operations are best carried out below 100°F. Although skimmers and diatomaceous earth precoat filters can operate at higher temperatures (e.g., 175°F), the temperature limit is usually based on the temperature at which silica dissolves. Incidentally, the typical optimum pH of water in a boiler is close to 9 to retard boiler tube corrosion (see Fig. 22.5).

22.3.2 Coordinated Phosphate-pH Boiler Water Treatment

Liz described an archaic form of boiler feedwater treatment that had saved the day for the utility plant in the refinery where she had been working when the refinery demineralization plant froze one winter resulting in seriously restricted flow and a great shortage of boiler feedwater. The technique and chemistry is described by a paper written by George Gibson (Betz)¹; although according to Mr. Gibson, it was apparently first introduced in 1942 by SF Whirl & TE Purcell of the Duquesne Light Company.

Liz says that they were able to treat raw city water and add it to the treated water for boiler feedwater simply by following the operating guidelines set by the coordinated phosphate-pH control chart. It works by causing hardness salts to be precipitated out as free-flowing sludge (calcium as phosphate, magnesium as hydroxide or silicate). This is achieved by continual additions of sodium phosphate and maintenance of pH within guidelines as set by the coordinated phosphate-pH control operating envelope which depends on the operating pressure of the boiler for which the feedwater is required. The alkalinity pH is adjusted by additions of sodium carbonate or sodium hydroxide.

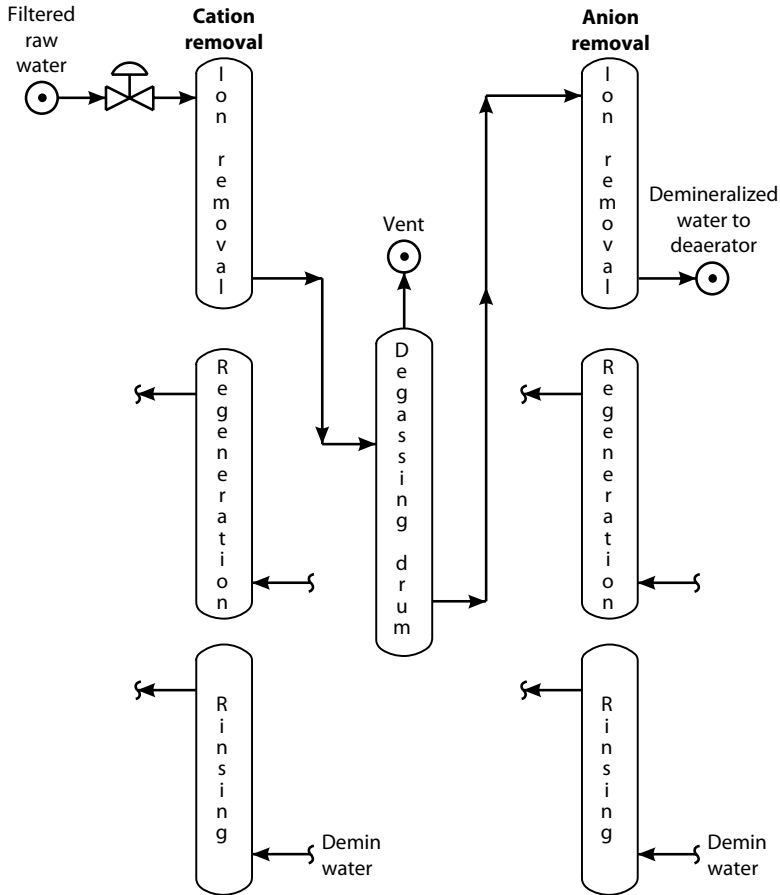


FIGURE 22.5 Demineralization plant used to minimize silicates and carbonates in plant steam.

22.3.3 Demineralization of BFW

The main advantages of cation-anion ion exchanger resin demineralization over phosphate treating or hot-lime softening of boiler feed-water is the superior reduction of silicates in steam. Silicates deposited on steam turbine blades may lead to a loss of turbine efficiency and vibration, and also stress corrosion, and sudden failure of the blades.

Such a “demin” plant consists of a cation exchanger resin bed followed by an anion exchanger resin bed. These resins look like small yellowish plastic beads and remind me of fish eggs (see Fig. 22.5).

The raw water flows downward through the cation exchange resin bed, where the metallic ions are removed. Then, in a typical installation, the partly treated water flows into a large degassing vessel.

Next, the water flows downward through the anion exchanger resin bed, where the silicates and carbonates and other such anions are removed.

Periodically, the resins have to be regenerated to strip out the absorbed ions. The regeneration solution (dilute acid for a cation bed, dilute alkali for an anion bed) flows in an upward direction. This is followed by a rinse step using treated water. Completion of the rinse is typically monitored by conductivity probes. The arrangement in Fig. 22.5 shows the operation of these resin beds in pairs. Basically, this means that six vessels are in service at a given time, but each pair is at a different stage in the cycle:

- Two absorbing ions (anions and cations)
- Two in regeneration mode
- Two in rinsing mode

It is important to avoid stirring up the resin during the regeneration step. The main objective of the regeneration step is to leave the purest resin at the bottom of the vessel. This will then “polish” the downflowing water when the bed is returned to its ion exchange service. A slight back-pressure, using air, is typically employed at the top of the beds to hold the little beads in place during regeneration, and works best after first draining the bed down by about 1 ft below the top of the resin bed.

It is also possible to lose the beads from the ion exchange vessels during the regeneration and washing steps, if the resin beds are not “held down” properly during this step.

Improper demineralization will also leave carbonates in the BFW. These carbonates will break down in the boiler to produce CO_2 . The CO_2 contaminates the steam supply to reboilers, where it is trapped. The CO_2 accumulates and dissolves in the steam condensate to produce corrosive carbonic acid, resulting in tube failure in the reboiler tube bundle.

22.4 Effect of Air Preheat on Boiler Capacity

One of the first questions that my coauthor, Liz, asked me at one of my seminars was:

“My utility plant operators have observed that when they preheat their boiler combustion air from 20°C to 220°C, steam production drops by around 3 or 4 percent. They are forced to reduce the firing rate, because the radiant section of the boiler starts to overheat. With the radiant section just as hot with or without the air preheat, I would think that steam production would remain constant, so why would it drop off?”

The answer to Liz’s question was that air preheat requires less fuel gas firing, and thus, less flue gas and hence less heat pickup in

the convective section. While the radiant section heat transfer rate will remain constant, the heat transfer rate in the convective section will drop in proportion to the reduced fuel gas fired. A 200°C (220°C–20°C) of air preheat will save 10 percent of the fuel and thus reduce the flue gas flow by the same 10 percent. As the convective section duty is typically 30 percent of the total boiler duty, a 10 percent reduction in the flue gas rate will reduce total boiler steam production by:

$$30\% \times 10\% = 3\%$$

Of course, I reminded Liz that she was also saving 10 percent of the fuel gas consumed for each pound of steam production, so perhaps the 3 percent boiler capacity reduction was acceptable.

“Liz,” I concluded, “That 3 percent is the price we have to pay for energy savings. Combustion air preheat in boilers, which are limited by excessive radiant section temperature or hot tubes, will always lose steam production capacity due to the air preheater.”

22.4.1 Adjusting Excess O₂

At the Tenneco Oil Refinery in New Orleans, I first learned how to optimize the excess oxygen content of the flue gas from a boiler. The fired boiler was equipped with a forced-draft (FD) fan. Due to mechanical deficiencies of the FD blower, the combustion air supply was limited, but constant. The boiler produced 650-psig steam, which was in short supply in the refinery. Hence, the boiler was operated to maximize steam production.

The utility plant operators would routinely adjust the fuel gas supply to maximize steam production. There was an accurate oxygen analyzer on the flue gas stack. It typically recorded “SEVERAL” percent oxygen in the boiler’s stack. However, this O₂ value was ignored, and correctly so, by the operators. They simply adjusted the fuel-gas rate, which had a variable heating value, to maximize steam production with a fixed amount of combustion air.

If, on the other hand, they had been trying to produce a fixed amount of steam, then they would have adjusted the air flow to minimize fuel-gas consumption in the boiler.

The reader will note that I emphasized the word “SEVERAL,” rather than stating a target O₂ range in this example. There is no such range. The optimum O₂ in the stack is whatever it is to maximize steam production. Which means, the O₂ analyzer served no function in the control of this boiler. And that statement is, in general, true.

22.5 Deaerator Operation

Being born on Earth, I grew up unaware of the true nature of oxygen. Submerged in a sea of N₂ and O₂, I was ignorant of how unique oxygen is in the universe as a whole. Except on our home planet, oxygen does

not exist in its free state. It's combined with H_2 , carbon, or sulfur. As a young process engineer, I was surprised to learn that oxygen is a terribly corrosive element. It must be precluded from most process environments, otherwise product degradation and corrosion will result.

In particular, all O_2 must be removed from BFW. This is done either with a chemical O_2 scavenger or with steam stripping. Stripping air out of BFW is relatively inexpensive compared to the use of chemicals.

A deaerator (Fig. 22.6) is used to remove oxygen. The more efficient the deaerator, the less scavenger chemical treatment is required. Residual O_2 in the deaerator effluent BFW is determined by a field titration kit, and the oxygen scavenger chemical addition rate is adjusted accordingly.

The steam and cold deaerator makeup water mix in the small stripping section shown in Fig. 22.6. Most of the steam condenses (typically 90 percent) to heat the cold water (120°F) to its boiling point temperature (250°F) at the deaerator operating pressure of 15 psig. The rest is vented to the atmosphere. This vented steam is being used to strip out the residual O_2 in the cold-water feed.

Note that there is sometimes a second, hotter feed to the deaerator, in addition to the cold-water feed to the stripping section. This secondary feed is steam condensate, which having already been steam, bypasses the stripping section. The exception to this generalization is steam condensate recovered from turbine exhaust vacuum

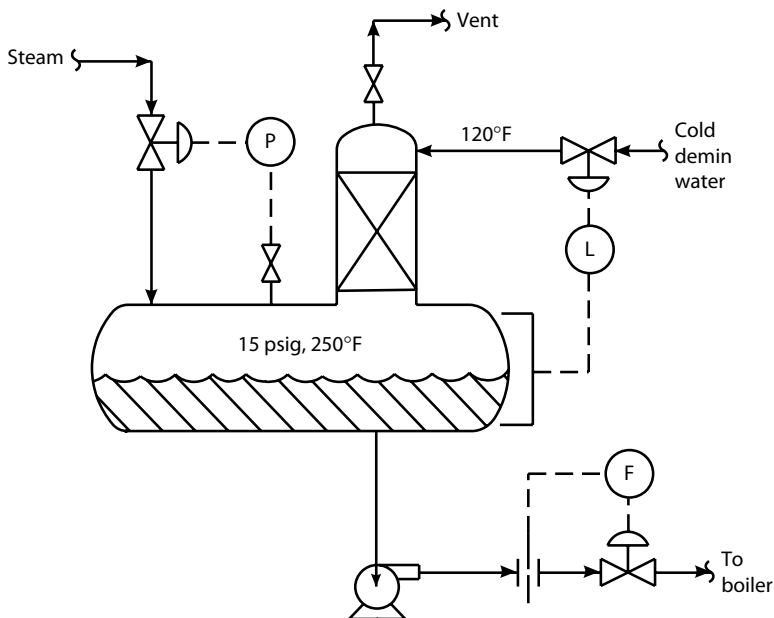


FIGURE 22.6 Deaerator is intended to remove O_2 from boiler feedwater.

surface condensers. Such condensate may have been in contact with air that has been drawn into the surface condenser shell and thus may have to be deaerated again. As noted above, liquid hydrocarbons must be removed by a clay polishing step to prevent oil accumulation in the deaerator, or laydown of a nonwetable film on boiler tubes.

In the event that there is too much residual oxygen in the deaerator effluent, the flow of vented steam from the top of the deaerator should be opened. However, often there is no vent valve, but just a fixed orifice vent. Or perhaps the supply steam control valve is 100 percent open. I had the latter problem at the Coastal Refinery in Aruba. The refinery was spending a small fortune on the oxygen scavenger chemical, pyridine. All I did was to lower the deaerator set point pressure from 18 to 13 psig, and the residual oxygen in the effluent BFW dropped by an order of magnitude.

I still recall how the formerly friendly chemical sales vendor suddenly became cold and hostile. And then canceled his luncheon invitation. It was worth the loss of the free lunch, however, to save the oxygen scavenger chemical costs.

As I've described in my book, *Troubleshooting Process Plant Control*, any time the steam inlet control valve becomes fully open, the deaerator temperature, pressure, and water level will become unstable and the deaerator stripping section will flood. Flood in the sense that boiling water may erupt from the atmospheric vent.

22.6 Boiler Feedwater Preheat

The optimum feed temperature to a deaerator is the boiling point of water at the deaerator pressure. Typically, 230°F to 240°F. However, in many of the plants, BFW to the deaerator is heated to 160°F to 180°F by heat exchange with process pumparound or product streams.

As a young engineer in the 1960s, I had learned that cooling water should not be heated above 125°F to 135°F to avoid laydown of hardness deposits inside the tubes of water-cooled exchangers. What I had failed to grasp was that once water has been through a demin plant, there is no restriction on how high a temperature that water can be heated.

The effluent from the deaerator (230°F) will then typically flow to the boiler's convective section, "ECONOMIZER," zone. This usually consists of four to six rows of tubes at the top of the convective section, where the BFW is heated to 350°F to 450°F. If the boiler is firing high-sulfur fuel oil (2 to 4 percent sulfur), this may be a problem. The dew point of H_2SO_4 in the flue gas may easily be as high as 450°F.

The 230°F water inlet may cause the flue gas temperature to drop below 450°F. This will precipitate a weak H_2SO_4 mist, which will aggressively corrode the fins or studs of the economizer tube bank. If low sulfur (less than 100 ppm H_2S is being fired), H_2SO_4 corrosion should not be a problem.

For boilers firing fuel oil (bunker), it's best to preheat the BFW externally to perhaps 350°F to protect the boiler's economizer convective tubes from corrosion. If a small amount of vaporization occurs in the preheat exchanger, this should not compromise the operation of the economizer, provided that there is only a single inlet to the economizer section.

22.7 Boiler Thermal Efficiency

To estimate the efficiency of fuel to steam, three factors are involved.

1. Approximately, for each extra 360°F or 200°C in stack temperature, boiler efficiency drops by 10 percent.
2. Approximately, for each extra percent of stack O₂, boiler efficiency drops by 1 percent.
3. For small package boilers (50,000 lb/hr), boiler external ambient heat loss is approximately 2 percent.

Thus, for a small boiler with a 600°F stack, 5 percent O₂ in the flue gas, boiler efficiency is:

$$100\% - 15\% - 5\% - 2\% = 78\%$$

Of course, the overall boiler plant efficiency will be lower due to:

- Steam leaks
- Power to drive FD and ID fans
- Power to drive BFW pumps
- Steam tracing for freeze protection
- Energy consumed in demin and deaerator
- Power to drive raw water and condensate recovery pumps
- Plant lighting and instrumentation

Finally, did you know that for each Btu worth of electricity, 3 Btu worth of steam are required?

22.8 Sloped Demister

On many boiler steam drums, there is a sloped demister to suppress entrainment and hence improve steam quality. However, if this demister becomes partly fouled with hardness deposits, it probably does more harm than good. In such cases, I would probably discard the demister rather than replace it. If you feel you have been doing a good job on level control in the boiler, but steam quality is still poor, you should consider this possibility.

Entrainment of high TDS boiler feedwater into steam from a boiler is promoted by excessive pH. Caustic is often added to the feedwater to reduce corrosion. However, if the pH increases much above 10.5, then the boiler tends to foam and hardness deposits are carried over into the steam. Hardness deposits degrade the performance of steam ejectors, steam superheating coils, and steam turbines.

Boiler Nomenclature

- Mud drum—Bottom part of boiler using thermosyphon water circulation.
- Steam drum—Where water and steam are separated.
- FD fan (also called F.D. Blower)—Supplies combustion air into a forced draft reboiler.
- Waste-heat boiler—Derives heat from convective section or process pumparound or product streams.
- Demin plant—Boiler feedwater is treated using ion exchange resin in a demineralization plant.
- Hot-lime softening—An older method to reduce the mineral content of boiler feedwater.
- Coordinated phosphate-pH control—Old method of treating water to reduce mineral content in BFW, causing precipitation of hardness salts as free-flowing sludge, used for power plant boilers.
- Deaerator—Used to minimize O₂ content of BFW by steam stripping.
- Oxygen scavenger—Chemical used to remove residual O₂ after BFW has been through deaerator.
- Condensate—The water recovered for reuse after steam has been condensed.
- Condensate polisher—Clay diatomaceous earth bed or filter used to remove oil from steam condensate.
- Silicates—Minerals in steam that plate out on steam turbine blades.
- Steam quality—A measure of the moisture content of steam.
- Saturated steam—Cooling steam from its saturation temperature, at constant pressure, causes 100 percent condensation of saturated steam.
- TDS—Total Dissolved solids.
- Blowdown—Draining several percent water from boiler to control TDS content of steam.
- Intermittent blowdown—Used to control suspended solids from mud drum of power plant boiler.
- Superheat—Heat added to saturated steam.
- Economizer—Used to preheat cold BFW.

References

1. George Gibson "The Basics of Phosphate - PH Boiler Water Treatment," Betz Laboratories, Inc. Power Engineering, February 1978.
2. British Standards Institution, "Recommendations for Treatment of Water for Land Boilers," BSI BS 2486:1978.

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CHAPTER 23

Vacuum Systems: Steam Jet Ejectors

The converging-diverging steam jet is a startlingly complex device. Not only is the theory of operation rather weird, but the jets are subject to a wide range of odd, poorly understood, and never reported malfunctions. For all these reasons, I dearly love to retrofit and troubleshoot steam jet systems.

Steam jets have been around for a long time. They have just as ancient an origin as do steam-driven reciprocating pumps. They were used on early steam engines to pull a vacuum on the now-archaic barometric condenser. More recently, they were used to develop vacuums in such services as

- Surface condensers that condense the exhaust steam from steam turbines
- Petroleum refinery crude-residue vacuum towers
- Flash evaporators used to produce concentrated orange juice

Steam jets are also employed to recompress low-pressure steam to a higher-pressure steam. Jets are sometimes used to compress low-pressure hydrocarbon vapors with higher-pressure hydrocarbon gas (instead of steam). They are really wonderful and versatile machines.

23.1 Theory of Operation

The converging-diverging steam jet is rather like a two-stage compressor, but with no moving parts. A simplified drawing of such a steam jet is shown in Fig. 23.1. High-pressure motive steam enters through a steam nozzle. As the steam flows through this nozzle, its velocity greatly increases. But why? Where is the steam going in such a hurry? Well, it is going to a condenser. The condenser will condense the steam at a low temperature and low pressure. It will condense the

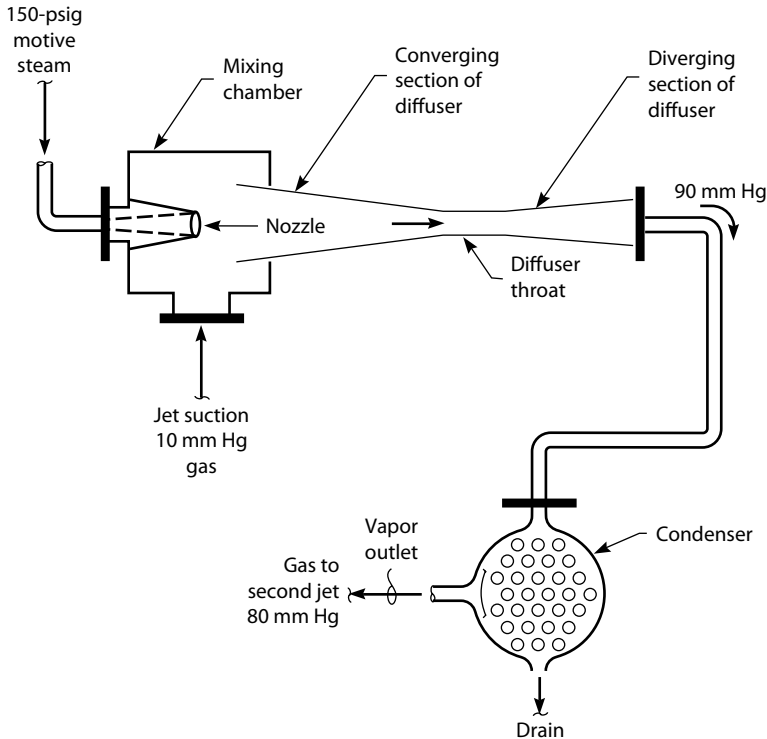


FIGURE 23.1 A converging-diverging jet.

steam quickly. The steam accelerates toward the cold surface of the tubes in the condenser, where its large volume will disappear as the steam turns to water.

The motive steam accelerates to such a great velocity that it can greatly exceed the speed of sound (that is, exceed sonic velocity). This tremendous increase in velocity of the steam represents a tremendous increase in the kinetic energy of the steam. The source of this kinetic energy is the temperature and the pressure of the steam.

The conversion of pressure to velocity is a rather common, everyday phenomenon. Remember Hurricane Opal, which struck the Florida Panhandle in October 1995? It had peak winds of 145 miles per hour. The pressure in the eye of the hurricane was reported at 27 inches Hg. A portion of the kinetic energy of the wind in a hurricane is derived from the barometric pressure of the atmosphere. The lift that helps an airplane fly is also a result of the conversion of barometric pressure to velocity. Because of the shape of the wing, the air passes across the top of the wing at a higher speed than it

does below the wing. The energy to accelerate the air flowing across the top of the wing comes from the barometric pressure of the atmosphere. Hence, the air pressure above the wing is reduced below the air pressure underneath the wing.

Similarly, as the high-velocity steam enters the mixing chamber shown in Fig. 23.1, it produces an extremely low pressure. The gas flows from the jet suction nozzle and into the low-pressure mixing chamber. The gas flows into the mixing chamber because there is a very low pressure in the mixing chamber.

The rest of the jet is used to boost the gas from the mixing chamber up to the higher pressure in the condenser. This is done in two compression steps: converging and diverging.

23.2 Converging and Diverging Compression

23.2.1 Converging Compression—Sonic Boost

When an airplane exceeds the speed of sound, we say that it breaks the sound barrier. In so doing, it generates a sonic wave or pressure wavefront. When steam and gas flow into the converging section of the jet diffuser shown in Fig. 23.1, the same thing happens. The gradually converging sides of the diffuser increase the velocity of the steam and gas as the vapor enters the diffuser throat above the speed of sound. This creates a pressure wavefront, or *sonic boost*. This sonic boost will multiply the pressure of the flowing steam and gas by a factor of perhaps 3 or 4.

Note something really important at this point. If for any reason the velocity of the steam and gas falls below the speed of sound in the diffuser throat, the sonic pressure boost would entirely disappear.

23.2.2 Diverging Compression—Velocity Boost

As the steam and gas leave the diffuser throat, the flow then enters the gradually diverging sides of the diffuser. The velocity of the steam and gas is reduced. The kinetic energy of the flowing stream is partially converted to pressure as the steam and gas slow down. This increase in pressure is called the *velocity boost*, which will multiply the pressure of the steam and gas by a factor of 2 or 3.

While smaller than the sonic boost, the velocity boost is more reliable. Even though the velocity in the diffuser throat in Fig. 23.1 falls well below the speed of sound, the increase in pressure in the diverging portion of the diffuser is only slightly reduced.

The overall pressure boost of a steam jet is obtained by multiplying the sonic boost effect times the velocity boost effect. The overall boost is called the jet's *compression ratio*.

23.3 Calculations, Performance Curves, and Other Measurements in Jet Systems

23.3.1 Vacuum Measurement

We have discussed the American system inches of mercury (inches Hg) in Chap. 10, “How Instruments Work.” Of more immediate interest is Table 23.1. To do any sort of vacuum calculation, we need to convert to the absolute system, in millimeters of mercury (mm Hg). Unfortunately, we also need to correct measurements made with an American-type (inches Hg) vacuum gauge for atmospheric pressure. You can interpolate between the two sets of data in Table 23.1, to correct for almost the entire range of typical atmospheric pressures.

We will need to use this table to calculate a jet’s compression ratio when we measure vacuum pressures with an American-type (inches Hg) gauge.

23.3.2 Compression Ratio

When considering the performance of a vacuum jet, we must first consider the jet’s overall compression ratio. To calculate a jet’s compression ratio:

1. Measure the jet’s suction pressure and convert to millimeters of mercury, as shown in Table 23.1.
2. Measure the jet’s discharge pressure and convert to millimeters of mercury.
3. Divide the discharge by the suction pressure. This is the compression ratio.

Actual atmospheric pressure = 29.97 inches Hg		Actual atmospheric pressure = 25.00 inches Hg	
inches Hg	mm Hg	inches Hg	mm Hg
0	760	0	635
5.0	633	5.0	508
10.0	506	10.0	381
15.0	379	15.0	254
20.0	252	20.0	126
25.0	125	25.0	0
29.97	0		

TABLE 23.1 Vacuum Measurement Systems

It is not uncommon to find a proper jet developing an 8:1 ratio. More typically, jets will develop a 3:1 or 4:1 compression ratio. Any jet with less than a 2:1 compression ratio has some sort of really serious problem, but not necessarily with the jet itself.

23.3.3 Jet Discharge Pressure

The jet suction pressure is a function of the following factors:

- The overall jet compression ratio.
- The jet discharge pressure, as shown in Fig. 23.1.

The jet discharge pressure is controlled by the downstream condenser pressure. The minimum condenser pressure corresponds to the condensing pressure of steam at the condenser's vapor outlet temperature. For example, let's say that the condensing pressure of pure steam at 120°F is 80 mm Hg. If the condenser vapor outlet temperature is 120°F, then the lowest pressure we could expect to measure at the condenser vapor outlet would be 80 mm Hg.

Let's further assume that the pressure drop from the jet discharge through the condenser discharge is 10 mm Hg. The jet discharge pressure would be 90 mm Hg. Let's also say that the sonic boost is equal to 3.60. The velocity boost is assumed to be equal to 2.5. The overall compression ratio is then

$$3.60 \times 2.5 = 9.0$$

The jet's suction pressure is then

$$(90 \text{ mm Hg})/9.0 = 10 \text{ mm Hg}$$

While I have seen steam jets develop compression ratios of 8:1 or 9:1, the majority of jets do not work nearly as well as that.

23.3.4 Multistage Jet Systems

A single jet that discharges to the atmosphere or to a condenser operating at atmospheric pressure is called a "hogging" jet. Let's assume that atmospheric pressure is 29.97 inches Hg or 760 mm Hg. Also, we will assume the jet is capable of an 8:1 compression ratio. The jet's suction pressure would be

$$\frac{(760 \text{ mm Hg})}{8.0} = 95 \text{ mm Hg}$$

Referring to Table 23.1, this would mean that the jet inlet pressure was 26.2 inches Hg on an American-type vacuum pressure gauge. This is about the best we could expect with a single-stage jet.

Figure 23.2 shows a three-stage steam jet system. Let's first calculate the overall compression ratio for the combined effect of all three jets.

Note that the atmospheric barometric pressure is 26 inches Hg (this is a chemical plant in Denver—the Mile High City). The first-stage jet

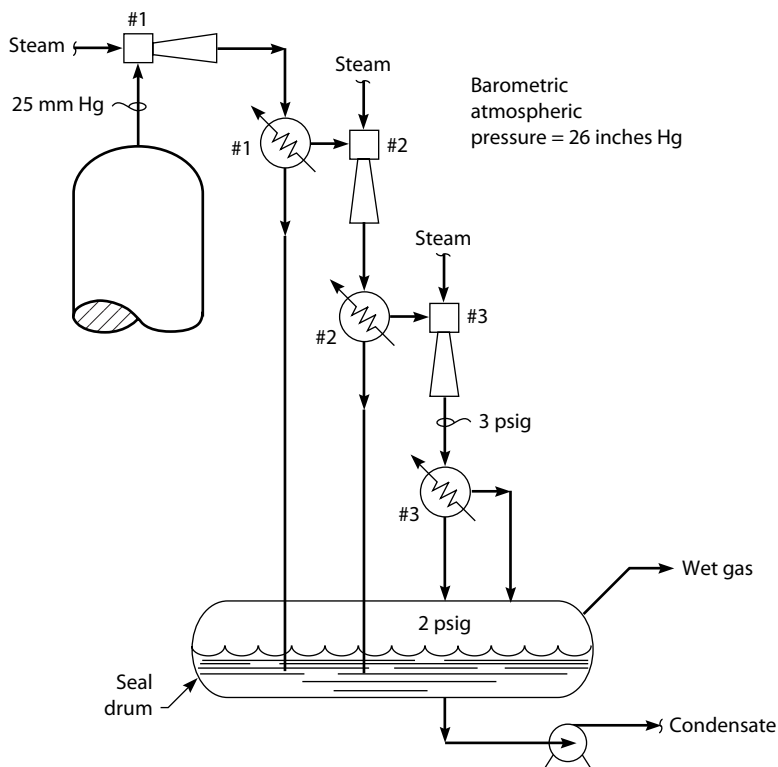


FIGURE 23.2 A three-stage jet system.

gas inlet pressure is 25 mm Hg. The third-stage jet discharge pressure is 3 psig. A good rule of thumb is

- 1 psi = 51 mm Hg

The actual barometric pressure can be converted to absolute mm Hg, using the following rule:

- 1 inch Hg = 25.4 mm Hg

Therefore, the barometric pressure in Denver is

- 26 inches Hg \times 25.4 = 660 mm Hg

The pressure at the third-stage jet discharge is then

- 660 mm Hg + (3 \times 51) mm Hg = 812 mm Hg

The overall compression ratio is

- $812 \div 25 = 32.5$

But what is the average compression ratio for each of the three jets? Well, let's assume that the pressure drop for the first two condensers in Fig. 23.2 is zero. Then let's remember that when compression stages

work in series, their compression ratios are multiplied together to calculate the overall compression ratio. Then the average compression ratio per jet is

$$(32.5)^{1/3} = 3.2$$

We take the cube root of the average compression ratio, because the three jets represent three compression stages working in series.

In this calculation, I have made a potentially serious omission. That is, I have ignored the pressure drop through the interstage condensers #1 and #2. Typically, these condensers have a shell-side ΔP of only a few mm of Hg. Thus their effect may be ignored. However, if these condensers are fouled, or if they are suffering from condensate backup submerging the bottom edge of their internal air baffles, then their ΔP might be 50+ mm of Hg. Such a large pressure loss must be included in the above compression ratio calculations.

For example:

- The inlet pressure to a first-stage jet is 20 mm Hg.
- The inlet pressure to a second-stage jet is 100 mm Hg.
- The ΔP across the interstage condenser is 60 mm Hg.
- Then the compression ratio of the first-stage jet is 100 mm Hg, plus 60 mm Hg, divided by 20 mm Hg.
- This is $(100 + 60) \div 20 = 8.0$.

23.3.5 Jet Performance Curves

I have rather implied up to now that a steam jet, depending on its mechanical condition, will develop a fixed compression ratio. This is not true. For one thing, the gas rate through the jet will influence its suction pressure. This is shown on the typical *jet performance curve* in Fig. 23.3, which is drawn for a constant discharge pressure of 100 mm Hg.

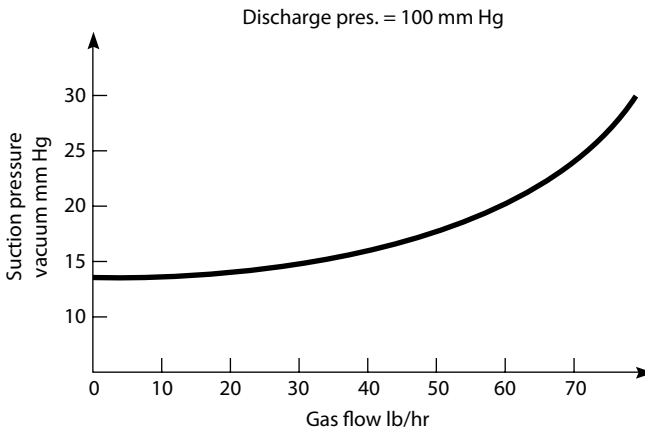


FIGURE 23.3 A typical vacuum steam jet performance curve.

Let me make a critical comment about this curve. Variations in the jet's discharge pressure may have a surprising (i.e., non-linear) effect on the jet's suction pressure. Sometimes, a small reduction in the discharge pressure will make a tremendous improvement in the suction pressure. The compression ratio might increase from 3:1 up to 7:1.

Sometimes, a very large reduction in the jet's discharge pressure will not alter its suction pressure at all. The compression ratio might decrease from 7:1 to 3:1. It all depends on something called the *critical-flow* characteristics of the jet. More on this subject in a moment.

23.3.6 Measuring Deep Vacuums

For any vacuums better than 120 mm Hg (or 25 inches Hg at sea level), an ordinary vacuum pressure gauge will not be accurate enough for technical purposes. An absolute mercury manometer, as shown in Fig. 23.4, is needed. All that is required to make this simple device is a length of glass tubing bent into a U-tube shape. One end is sealed and the other end is left open. Dry, clean mercury is then poured into the open end. The closed end of the U-tube is easily evacuated of air by tipping the glass U-tube on its side. A little jiggling will work out the last air bubble. The overall length of the tube will be about 8 inches. The mercury should wind up about 1 or 2 inches high in the open end of the U-tube.

To read the vacuum, the mercury level at the closed end must be pulled down by the vacuum even just a little below the top of the tube.

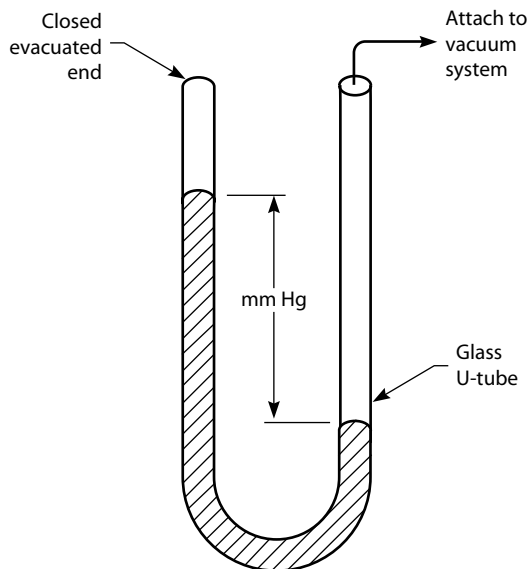


FIGURE 23.4 An absolute vacuum mercury manometer.

The difference in mercury levels between the closed and open ends of the U-tube is the precise mm Hg vacuum. Wet mercury will give completely wrong results.

23.3.7 Jet Malfunctions

Big Spring is located in the scrub desert of western Texas. Cold autumn mornings are followed by warm afternoons. The local refinery's cooling-water temperature follows this ambient-temperature trend. The vacuum tower in this refinery also seems to keep track of the time of the day.

At 6:00 A.M., the primary steam jet is running quietly and pulling a vacuum of 12 mm Hg. At 7:30 A.M., the jet begins to make infrequent surging sounds. It rather sounds as though the jet is slipping every 15 or 20 seconds. As the morning coolness fades, the surging becomes more frequent. The vacuum also begins to slip from 12 to 14 mm Hg. Then, at about 9:00 A.M., as the surges have become so frequent as to be almost continuous, the vacuum plunges to 23 mm Hg. By 10:00 A.M., the surging has stopped and the vacuum at the jet suction has stabilized at a poor 25 mm Hg. Many, if not most, operators of large vacuum jet systems have observed this problem—but what causes it?

23.3.8 Loss of Sonic Boost

Let's refer to Fig. 23.2. As the cooling water warms, the temperature of condenser #1 increases. This also increases the condensing pressure. This raises the discharge pressure of the primary jet (jet 1), as well as the pressure in the diffuser throat (see Fig. 23.1). Higher pressure will result in a smaller vapor volume. And a smaller vapor volume will cause a reduction of the velocity in the diffuser throat.

The lower velocity in the throat does not affect the jet's performance, as long as the velocity remains above the speed of sound. If the velocity in the throat falls below the speed of sound, we say that the jet has been forced out of critical flow. The sonic pressure boost is lost. As soon as the sonic boost is lost, the pressure in the vacuum tower suddenly increases. This partly suppresses vapor flow from the vacuum tower. The reduced vapor flow slightly unloads condenser #1 and jet 2, shown in Fig. 23.2. This briefly draws down the discharge pressure from jet 1. The pressure in the diffuser throat declines. The diffuser throat velocity increases back to, or above, sonic velocity. Critical flow is restored, and so is the sonic boost. The compression ratio of the jet is restored, and the vacuum tower pressure is pulled down. This sucks more vapor out of the vacuum tower and increases the loads on condenser #1 and jet 2 (the secondary jet). The cycle is then repeated. Each of these cycles corresponds to the surging sound of the jet and the loss of its sonic boost.

As the cooling-water temperature rises, the sonic boost is lost more easily and more rapidly. The surging cycles increase in frequency

to 30 or 40 per minute. The vacuum tower pressure becomes higher and higher. Finally, the surges become so frequent that they blend together and disappear. The primary jet has now been totally forced out of critical flow. The sonic boost has been lost until the sun in Big Spring, Texas, sets and the desert cools. Surging then returns until the critical flow in the jet is restored, and the sonic boost is regained at about 9:00 P.M.

23.3.9 Restoring Critical Flow

Steam jets, especially the primary jets, are forced out of critical flow, most commonly because

- Inadequate capacity of the primary condenser (i.e., condenser #1 shown in Fig. 23.5).
- Overloading or poor performance of the second-stage jets.

The problem at Big Spring was rather typical. The two parallel, second-stage jets were not working as a team. Jet A in Fig. 23.5 was a real strong worker. Jet B was a loafer. It is rather like running two centrifugal pumps in parallel. Unless both pumps can develop about the same feet of head, the strong pump takes all the flow, and the weak pump is damaged by internal recirculation. In the case of jets working in parallel, the strong jet takes all the gas flow from the upstream condenser. Furthermore, the strong jet sucks motive steam out of the mixing chamber (see Fig. 23.1). As you can see in Fig. 23.5, the suction temperature of jet A is 20°F hotter than the condenser #1 outlet temperature. This could happen only if the flow of vapor in

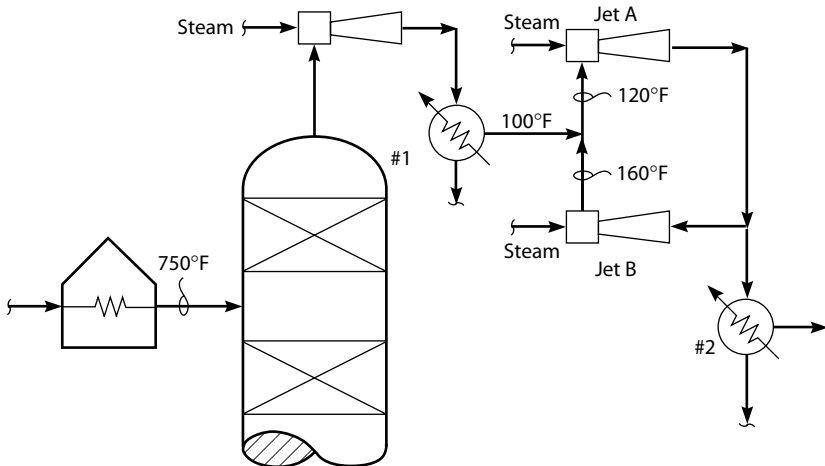


FIGURE 23.5 A malfunctioning secondary jet.

the jet B suction line were backward. Blocking-in jet B stopped the primary jet from surging until the more intense heat of the late afternoon.

When I block-in one ejector that is running in parallel with one or two other jets, I always try to remember to close the large process inlet valve first. If you close the smaller motive steam valve first, then the discharge flow from the other parallel ejectors will recycle back through the jet you are removing from service. The vacuum will suddenly break. So block-in the steam valve second, and the discharge valve from the jet third. I've made the error of closing the steam valve prematurely, so you can be guided by my errors.

23.3.10 Calculating Sonic Velocity

I have discussed in this chapter the effect of reaching sonic velocity, also called the critical flow velocity or choke flow.

To calculate sonic velocity, VS in feet per second:

$$VS = 223 (K \cdot T \div MW)^{1/2}$$

where $T = ^\circ\text{F} + 460 = ^\circ\text{R}$

MW = Molecular weight, pounds

K = Ratio of specific heats ($C_p \div C_v$)

K for ethane is 1.2 K for air and hydrogen is 1.4. For higher molecular weight components, assume K is 1.1.

Sonic velocity is also referred to as choke flow. One characteristic of choke flow is that reducing the downstream pressure does not significantly increase the flow through the choke point. Increasing the upstream pressure increases the flow in a linear proportion to the absolute increase in pressure.

A vapor-liquid mixture developing sonic velocity in a pipe will cause the liquid to atomize. The tiny droplets of liquid will then be difficult to settle out in a downstream separator vessel. This is often a cause for entrainment in vacuum tower feed nozzles.

23.3.11 Effect of Gas Rate

The most effective operating change to restore a jet to its critical-flow mode is to reduce gas flow. As I have described, this reduces the primary jet's (jet 1's) discharge pressure. Also, less of the energy of the motive steam is expended in accelerating the reduced gas flow. Hence, the steam enters the diffuser throat with greater kinetic energy. This also helps, along with the lower discharge pressure, in restoring critical flow and the jet's sonic boost.

To reduce the gas flow from the vacuum tower, shown in Fig. 23.5, I cut the heater outlet temperature from 750 to 742°F. This reduced thermal cracking in the vacuum heater and the consequent production of cracked gas. The pressure in the vacuum tower dropped from 21 to 12 mm Hg, and production of valuable heavy gas oil

from the vacuum tower-bottom residue increased by 20 percent. I can still recall the warm afternoon sunshine on my face as I signaled to the control-room operator to drop off that final degree of heater outlet temperature. I can still hear the last surge dying away as the primary jet recovered from its long illness. And as the jet began its steady, full-throated roar, I knew it was running in its proper critical-flow mode.

23.3.12 Reducing Primary-Jet Discharge Pressure

Let's say that a jet is already in its critical-flow mode. It is already benefiting from both the sonic boost and the velocity boost. What, then, will be the effect of a reduction in the jet's discharge pressure on the jet's suction pressure? Answer—not very much. If a reduction in discharge pressure is made on a jet that is *not* working in its critical mode, there will always be some benefit.

But if the jet is already in critical flow, reducing the pressure downstream of the diffuser throat cannot significantly raise the flow of gas into the diffuser throat. I know, I've tried. Twice I have added a third-stage jet to an existing two-stage jet system. The discharge pressure from the second-stage jet dropped by 500 mm Hg. The discharge pressure of the first-stage jet dropped by 160 mm Hg. The suction pressure to the first-stage jet dropped by perhaps 2 mm Hg.

The critical discharge pressure for each jet is determined experimentally by the manufacturer. It is usually noted on the jet specification sheet. My experience indicates that exceeding this critical jet discharge pressure by the smallest amount will force the jet out of critical flow. Or, the way I see it, will cause the jet to suddenly surge a few times and then lose its sonic boost.

The inverse is not always true. A jet with less than its critical discharge pressure may not pick up its sonic boost. Operationally, this business of gaining or losing sonic boost is a rather dramatic effect.

23.3.13 Condensate Backup

The jet in Fig. 23.6 has a small calculated compression ratio of about $(180 \text{ mm Hg})/(150 \text{ mm Hg}) = 1.20$. This extremely low compression ratio does not indicate any sort of jet malfunction. The high jet suction pressure is caused by the 140°F precondenser outlet temperature. The vapor pressure of water at 140°F is 150 mm Hg. There is a large amount of process steam flowing into the precondenser. The lowest possible pressure that the precondenser can operate at and still condense the process steam is 150 mm Hg. As the jet sucks harder, it just pulls a few more pounds of water out of the precondenser, without altering the precondenser's pressure.

The problem with the precondenser is condensate backup. Something, perhaps a partially plugged drain line, is restricting

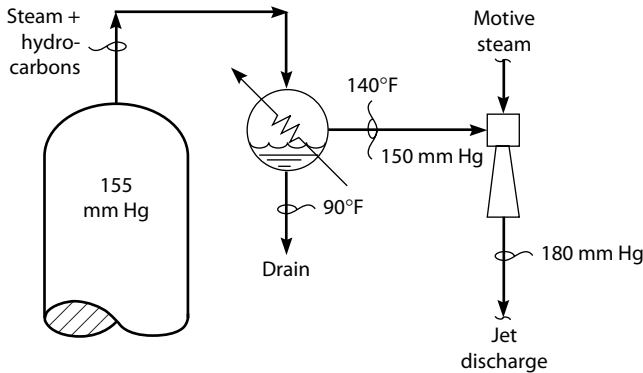


FIGURE 23.6 Condensate backup limits vacuum.

condensate flow. As the condensate backs up, it reduces the surface area of the condenser exposed to the condensing process steam. This makes it more difficult for the process steam to condense. The condensate backup also subcools the condensate. The net result is that the precondenser vapor outlet temperature goes up and the precondenser liquid outlet temperature goes down.

Recently, on a job in Arkansas City, I was able to force a jet to surge and lose its sonic boost simply by raising the condensate level in its downstream condenser by just 6 inches. Lowering the level drew down the jet's discharge pressure by a few millimeters of mercury and restored it to critical flow.

Condensate backup is also caused by a large number of problems associated with the barometric drain line, and especially with the seal drum, shown in Fig. 23.2:

- Air leak in drain line. This caused local chilling of the line near the leak. Likely, the intrusive air was mixing with the light liquid hydrocarbon in the line and promoting autorefrigeration.
- Wax plugging in the drain line. Gas oil entrainment is often quite waxy. The cure is to steam-out the drain line.
- Sludge buildup in the seal drum. The bottom of the seal legs were only 4 inch above the bottom of the seal drum. The seal drum overflow baffle was 4 ft high. I had the drum cleared and cut at 18 inch off the seal legs.
- Biological corrosion inside the seal drum. The bacteria metabolized H_2S and iron in the warm seal drum aqueous phase to form the sludge. If the internal seal drum legs are made of carbon steel [should use 316 (L) stainless steel], the internal seal drum legs will hole-through and unseal the barometric legs.

Running at very high liquid levels to submerge the hole is the temporary cure.

- False seal drum level indication. The level had been showing a dead-steady 80 percent level for months. Actual level when I blew-down the liquid level taps was 100+ percent.
- Too much naphtha in the vacuum tower feed made it impossible for the vacuum condensate seal drum slop pumps to keep the level under control. Turning on the spare pump did not help, as both pumps were operating on the flat portion of their performance curves.

I have written a separate book pertaining to a wide variety of vacuum system problems: *Troubleshooting Vacuum Systems* (Wiley, 2012). And my chapter on steam surface condensers also deals with pertinent vacuum system malfunctions.

23.3.14 Jet Freeze-Up

There is another type of jet surging, which is caused by the motive steam turning to ice. How is this possible? Certainly, steam cannot turn to ice inside the jet? But it can and does.

The jet system was in Mobile, Alabama. The symptoms of the problem were

- Extremely poor performance of the primary jet.
- The poor performance was constant, regardless of the cooling-water temperature.
- The jet would roar along in a normal fashion, and then go extraordinarily quiet for 10 to 15 seconds.
- The outside of the mixing chamber would chill to exactly 32°F during those times when the jet was quiet. This happened even though it was a bright, warm, sunny day in Mobile.

Incidentally, the way to measure surface temperatures in the field is with an infrared handheld thermometer. The response time of these thermometers is infinitely faster than in old-style, contact thermometers. You can buy one from any good instrument catalog¹ for a few hundred dollars.

Naturally, if the jet freezes, steam flow will stop. The jet will be quiet, and its compression ratio will be nil. But what causes the steam to turn to ice? Well, a number of factors extract heat from the steam:

- When any vapor expands, due to a pressure reduction (other than H_2 and CO_2), it cools off. This is called a *Joule–Thomson expansion*. The reduction in temperature of the steam is called

a *reduction in sensible-heat content*. The sensible heat of the steam is converted to latent heat of condensation. Does this mean that the latent heat of condensation of 10-psig steam is much higher than that of 450-psig steam? Let's see:

- Latent heat of condensation of saturated 10-psig steam = 980 Btu/lb.
- Latent heat of condensation of saturated 450-psig steam = 780 Btu/lb.
- When the velocity of a vapor increases, some of the increase in kinetic energy is extracted from the sensible heat of the vapor.
- The steam also gives up some of its energy to provide increased momentum to the gas flowing into the jet suction nozzle (see Fig. 23.1).

If the motive steam were dry, these factors would reduce the 150-psig motive-steam saturated temperature from 350°F to about 100°F. But the motive steam in Mobile was not dry. It had partly condensed in the steam supply line to the jet. If steam is wet and contains liquid water, the water will flash to steam when the steam pressure is suddenly reduced to a vacuum. But the heat of vaporization comes from the sensible-heat content of the steam. If the steam contains 10 percent moisture, it will chill by 180°F on flashing. This implies that we could have the wet, motive-steam temperature dropping to -80°F as it enters the jet's mixing chamber. But, of course, the steam will turn to ice when its temperature drops to 32°F. The ice blocks the flow of steam. As the steam velocity slows, the jet warms and melts the frozen steam, and the steam flow is restored.

23.3.15 Wet Steam

The problem in Mobile was resolved by installing a small steam filter on the steam line to the jet. This filter extracts moisture from the steam and blows it out through a steam trap.

But wet steam is bad for a jet, even when it does not cause the jet to freeze. Mainly, wet steam causes erosion of the steam inlet nozzle. Erosion of this nozzle is the main reason why jets undergo mechanical deterioration. As the nozzle erodes, it allows more steam to pass through into the diffuser. The diameter of the diffuser is designed to operate with a certain steam flow. If that design steam flow is exceeded, the diffuser operation suffers. Also, the downstream condenser pressure will also increase.

An eroded steam nozzle shows no obvious sign of damage. The erosion is quite uniform and the nozzle interior is smooth. The inner diameter of the jet must be checked carefully with a micrometer. Growth in diameter of just 5 to 10 percent is significant. The nozzle is intended to be replaced periodically, much like the impeller wear ring on a centrifugal pump.

23.3.15.1 Steam Ejector Temperature Profile

The motive steam, as it flows through the ejector, passes through three distinct portions of the ejector body:

- First—Steam nozzle
- Second—Converging section of the ejector
- Third—Diverging section of the ejector

The function of each of these three sections is:

- First—In the steam nozzle, the temperature (enthalpy) of the steam is converted to velocity (kinetic energy).
- Second—In the converging section, the velocity of the steam is used to compress the off-gas, air leaks, and the motive steam itself, due to the combined vapor flow exceeding critical velocity (the sonic boost).
- Third—In the diverging section, the velocity of the steam is used to compress the off-gas, air leaks, and the motive steam, due to the reduction in the velocity of the combined vapor flow (the velocity boost).

The temperature profile is observed by monitoring skin temperatures with an infrared temperature gun. When I do this, here is what I've observed:

- First—If the motive steam is dry, saturated 150-psig, 360°F steam, then the temperature at the front portion of the ejector body will be around 100°F. The 360°F steam temperature has been converted into velocity in the steam nozzle.
- Second—If the inlet temperature at the converging section of the ejector is 100°F, then the temperature halfway along the ejector (at its throat or narrowest portion) will be about 200°F. The ejector has been heated by 100°F, due to the heat of compression of the flowing vapor provided by the sonic boost.
- Third—If the inlet temperature at the diverging section of the ejector is 200°F, then the temperature at the discharge of the jet will be about 250°F to 300°F. The ejector has been heated another 50°F to 100°F, due to the heat of compression of the flowing vapor provided by the velocity boost.

Sometimes the first portion of the ejector will be a lot colder than 100°F. That's because there are several percent of moisture in the motive steam, which is bad.

Sometimes the converging portion of the ejector will heat up by 10°F or 20°F, rather than by 100°F, as it should. That's because there is no compression taking place in the converging portion of the ejector, which is also bad. Lack of a significant temperature rise

(about 100°F) across the ejector's converging section is a positive indication that the ejector is failing to develop its sonic boost. This is the principal malfunction encountered in troubleshooting steam ejector performance.

The temperature profile I've just described will vary with:

- Vapor loads (air, cracked gas)
- Vapor composition (molecular weight)
- Ejector mechanical condition
- Steam conditions (moisture or superheat)
- Ejector discharge pressure, as controlled by the downstream condenser performance
- Suction pressure

I had observed a wide variety of temperature profiles of ejector bodies, and for many years I failed to understand the significance of my observations. After over 35 years of confusion, I decided to analyze the data using:

- My education in thermodynamics
- Mollier diagram for steam

I find it strange that the engineering principles that I was supposed to have learned when I was 20, I only realized how to actually apply when I was well past 60. But, now that I think about it, pretty much all of life is like that.

Summary: A small (10°F to 20°F) temperature rise across the converging section of a steam jet is a definite indication of the loss on the sonic boost and of a low ejector compression ratio. The overall temperature rise across the ejector's diffuser section should be 150°F to 200°F. I have written an entire book on this complex subject, *Troubleshooting Vacuum Systems* (Wiley, 2012).

23.3.16 Don't Know Why, But It Happens

I have learned a lot about how process equipment actually works by investigating comments such as "It may not make any sense to you, but that is what happens here." Process equipment always conforms to the principles of science, but we have to get to know which principle to apply.

Figure 23.7 shows an old vacuum tower in Aruba. The chief operator on this unit made the following statements:

- "The colder the vapor outlet temperature from the precondenser, the better the vacuum that the ejector could develop, because of reduced vapor flow to the jet." Agreed!

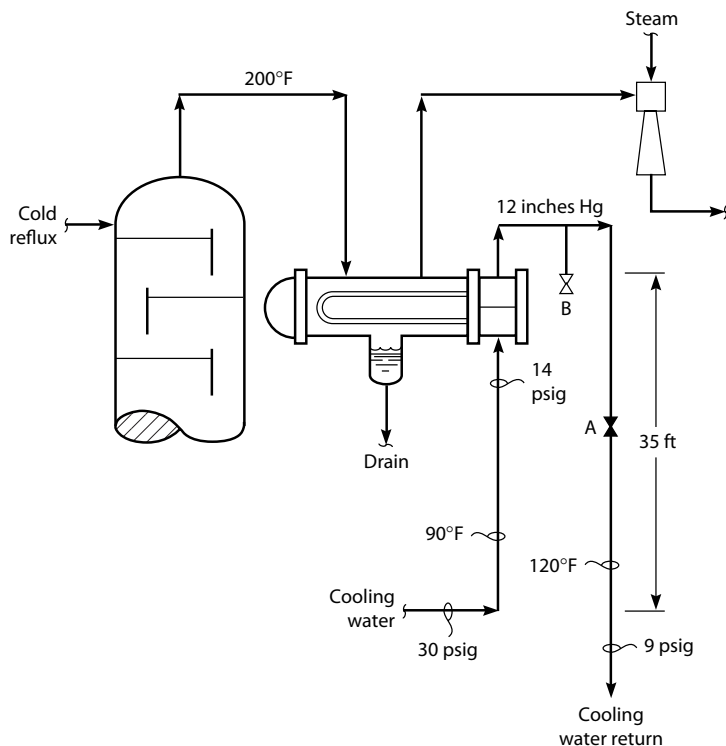


FIGURE 23.7 Air evolving from cooling water reduces water flow.

- “Increasing cooling-water flow to the precondenser decreases the vapor outlet temperature.” Agreed!
- “Closing the cooling-water outlet valve A, about three-fourths of the gate valve steam travel, increases cooling-water flow through the precondenser.” Nonsense!

Really, dear reader, how can closing a valve in a pipeline increase flow in that pipeline? It cannot, and it will not, and it did not. Yet, on the other hand, it is a really bad idea to disregard field observations made by experienced plant operators. So, let's take a closer look at Fig. 23.7.

First, I tried opening valve A. Just as the chief operator said, the cooling-water outlet temperature increased, proving that water flow was reduced. Next, I checked the pressure at bleeder B; it was 12 inch of Hg. The pressure was so low at this point because of

- The 20-psi ΔP of the cooling water as it flowed through the tube bundle.
- The 35 ft of elevation (about 15 psi of head pressure) that the water had to gain to climb to valve B. Of course, this 15 psi of

head loss was regained when the water flowed back down to the cooling-water return header.

The first idea I had was that the 120°F water would partially flash to steam at 12 inches Hg and the evolved vapor would restrict water flow. Wrong! The vapor pressure of water at 120°F is 26 inches Hg, not 12 inches Hg.

But cooling water is not just pure water. It is water that has been saturated with air in the cooling tower. Sure enough, when I calculated the amount of dissolved air that would flash out of water (which had been saturated with air at 90°F and atmospheric pressure), I found that a very large amount of air could flash out of solution. It was just as the chief operator had said. Opening valve A too much evolved large volumes of air in the second pass of the U-tube precondensers, shown in Fig. 23.7. Certainly, the total volumetric flow through the outlet of the condenser does increase, as valve A is opened. But the incremental flow is all air, and all that air does is choke off the cooling-water flow.

23.4 Optimum Vacuum Tower-Top Temperature

The chief operator also insisted that lowering the vacuum tower-top temperature too much would hurt the vacuum. But why? There is no doubt that the colder the tower-top temperature, the less the heat-duty load for the precondenser to absorb. Hence, cooling the vacuum tower-top temperature should, and did, reduce the precondenser vapor outlet temperature. This should have reduced the vapor load to the downstream jet. But it didn't. Here is why:

- The vapor components distilled overhead from the vacuum tower consisted of steam, cracked gas, and naphtha.
- The steam and naphtha vapors would pretty much totally condense in the precondenser, shown in Fig. 23.7.
- Some of the cracked gas would dissolve in the condensed naphtha. Most of the cracked gas would flow onto the jet.
- Increasing the tower-top temperature would distill over more pounds of naphtha.
- The extra condensed naphtha would dissolve more cracked gas.
- The reduced flow of cracked gas to the jet would unload the jet and permit it to develop a larger compression ratio.

Of course, if the vacuum tower-top temperature became too high, the increase in the precondenser vapor outlet temperature would increase the vapor pressure of water. This factor would then limit the minimum pressure in the precondenser.

In another case, a large volume of NH_3 was accidentally injected in the inlet to the condenser. The vacuum instantly improved. Why?

Well, the NH_3 reacted with the H_2S in the cracked gas to form NH_4HS (ammonium sulfide). This salt is very soluble in water; H_2S is not. The H_2S was effectively extracted from the cracked gas, and the downstream jet was thus unloaded and sucked harder.

Jets have been partially replaced by liquid ring-seal pumps. These are really positive-displacement compressors. The gas is squeezed between the vanes of the compressor's rotor and a pool of liquid in the compressor's case. Liquid ring-seal pumps are not interesting. They have no character. They are not as complex as steam jets. Anyway, I will discuss positive displacement compressors in later chapters.

23.5 Measurement of a Deep Vacuum without Mercury

Above 30 or 40 mm Hg of vacuum, an ordinary digital battery powered vacuum gauge is sufficient. However, only a vacuum mercury manometer is accurate in the 5 to 15 mm Hg range. As long as the mercury was dry, such a gauge gave excellent results. But few of my clients now permit the use of mercury in the field.

I have developed an alternate method that works almost as well as the vacuum mercury manometer. It's really the same idea (see Fig. 23.8). Fill the closed end of the tubing with baby oil and mineral oil. Make sure you get the last bubble of air out of the closed end of the tubing. Always use fresh oil to prevent moisture

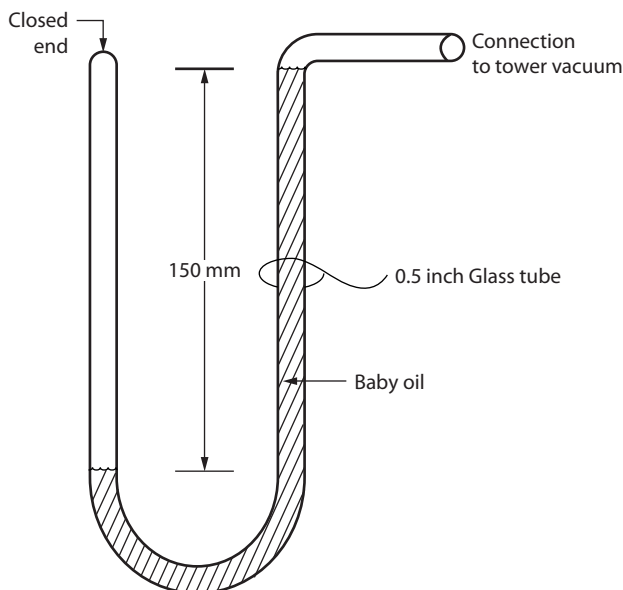


FIGURE 23.8 Oil-filled deep vacuum manometer.

contamination of the oil. Fill the entire glass U tube with oil. Connect the open end of the glass tubing to the vacuum pressure point being measured and open the connecting valve very slowly.

The specific gravity of baby oil is about 0.9 s.g. The specific gravity of mercury is 13.6. Mercury is 15 times denser than baby oil. Referring to Fig. 23.8, divide the 150 mm of elevation difference in the manometer legs by 15 to determine that the vacuum is 10 mm Hg. For vacuums above 40 mm Hg, the oil-filled vacuum manometer is not practical, as it would be about 30 inches in overall length.

Do not use lighter fluids than baby oil, as they will create a vapor pressure at the closed end of the glass U tube. This will result in a measurement of a better vacuum than really exists. Water is also too volatile at moderate ambient conditions for this purpose.

23.5.1 Troubleshooting Vacuum Systems

I have written an entire book on the subject of vacuum system malfunctions: Norman P. Lieberman, *Troubleshooting Vacuum Systems* (Wiley, 2012), ISBN 978-1-118-29034-7, www.wiley.com.

This book is based on my 48 years of field troubleshooting steam ejectors and surface condensers. The preceding chapter barely reflects the depth and complexity of this subject. However, if I had to select the most common problem with steam jets, I would choose steam hardness deposits that accumulate inside the steam nozzles. These deposits are a consequence of entrainment of water droplets in the steam from the boiler due to poor boiler feed water level control.

Reference

1. *Davis Instruments* (catalog), vol. 60, copyright 1995, Davis Instruments, 4701 Mount Hope Drive, Baltimore, MD 21215.

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CHAPTER 24

Steam Turbines

Use of Horsepower Valves and Correct Speed Control

A *steam turbine* is a machine with an ancient genealogy. It is a direct descendant of the overshot water wheel, used to kick off the Industrial Revolution in England, and the windmill still used in Portugal. Turbines are widely used in process plants to drive everything from 2-horsepower pumps to 20,000-hp centrifugal compressors. They are versatile machines, in that they are intrinsically variable-speed devices. Electric motors are intrinsically fixed-speed machines. It is true that there are a variety of ways to convert AC (alternating-current) motors to variable speed, but they are all expensive and complex.

There are two general types of steam turbines: extraction and condensing. The most common turbine with which the process operator comes into contact is used to spare an electric-motor-driven centrifugal pump. The three-phase, AC motors used in the United States and South America are either 1800 or 3600 rpm. The motors used in Europe are 1500 or 3000 rpm. Small steam turbines (20 to 500 hp) used to drive centrifugal pumps are rated for the same speed as the electric motors that are used in that particular service.

24.1 Principle of Operation and Calculations

Did you ever turn a bicycle over on its handlebars and squirt water from a garden hose at its front wheel? What causes the wheel to spin with such great speed? What is the property of water striking the spokes that causes the wheel to spin? Is it the pressure of the water hitting the spokes that spins the wheel? Certainly not! While the water pressure in the hose might be 40 psig, as the water discharges from the hose nozzle, its pressure surely falls to the atmospheric pressure in our garden. But what happens to the pressure of the water? It is converted to velocity!

It is the velocity of the water, then, that causes the wheel to spin. The velocity of the water is transferred to the spokes of the wheel.

What force of nature causes a windmill to spin? Answer—the wind (or the velocity of the air). What causes a steam turbine to spin? Answer—the velocity of the steam hitting the turbine wheel.

24.1.1 A Simple Steam Turbine

Figure 24.1 is a conceptual drawing of a simple topping steam turbine. An actual turbine does not look anything like this sketch. The 400-psig motive steam enters through a *governor speed-control* valve. This valve, sometimes called the *Woodward governor*, controls the flow of steam into the *steam chest*. If the turbine is running below its set speed, the governor valve opens.

It is really just like cruise control on your car. You select the speed you want to drive at. If the car is going slower than the speed you selected, its set speed, the gas pedal is depressed to bring the car up to your selected set speed. The governor speed-control valve is just like the gas pedal.

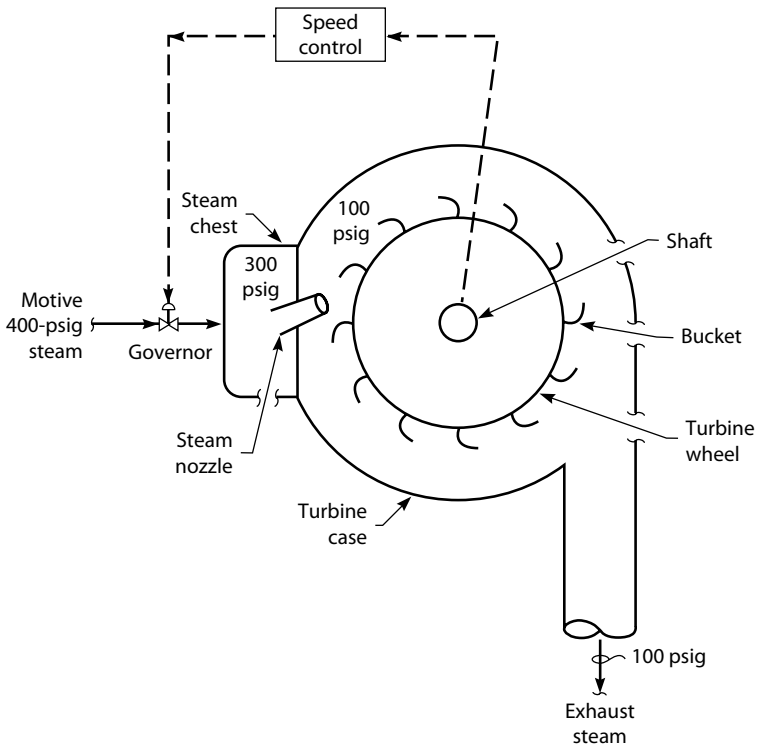


FIGURE 24.1 Conceptual sketch of a steam turbine.

The pressure drop across the governor control valve in Fig. 24.1 is 100 psi. This sort of pressure drop is called an *isoenthalpic expansion*—a term used in thermodynamics meaning that the heat content of the steam has been preserved during its reduction in pressure from 400 to 300 psig. It is also an *irreversible expansion*, meaning that the ability of the steam to do work has been reduced. As the steam passes through the governor, a substantial amount of its pressure is lost, because of friction in the governor valve.

The 300-psig steam next passes through the steam nozzle. This is an ordinary nozzle. It screws into a hole in the wall that separates the steam chest from the *turbine case*. The nozzle is shaped to efficiently convert the pressure of the 300-psig steam to velocity. The pressure of the steam, as soon as it escapes from the steam nozzle, is already the same as the exhaust steam pressure (100 psig).

All the pressure energy lost by the steam in expanding from 300 to 100 psig is converted to velocity. This is called an *isoentropic* or *reversible expansion*. The term *isoentropic* is a thermodynamic expression meaning that the *entropy* of the steam has not increased. The term *reversible* in this context means that I could take the high-velocity, 100-psig steam exhausting from the nozzle, run it backward through an identical nozzle, and convert the steam's kinetic energy back into 300-psig steam. An isoentropic expansion is also frictionless, in that the ability of the steam to do work is preserved.

If you are using your Mollier diagram, you will have noted that the temperature of the steam has dropped from 375°F to 220°F. The 220°F is the temperature of 17-psia saturated steam. That 155°F drop in the steam temperature represents 85 Btu/lb. The 85 British Thermal Units (Btu) has been converted into kinetic energy. Most of the kinetic energy of the steam, as it passes through the steam nozzle, is derived from the temperature of the expanding steam.

The high-velocity steam strikes the buckets around the rim of the turbine wheel shown in Fig. 24.1. Actually, these so-called buckets resemble blades or vanes. I imagine the term “bucket” is a carry-over from the days of the water wheel. The velocity of the steam is now transferred to the spinning turbine wheel. If the turbine is running below its set speed, the speed controller causes the governor speed-control valve to open to allow more steam to flow into the steam chest.

24.1.2 Calculating Work Available from Motive Steam

Dear engineering reader, please recover from your desk drawer your *steam tables*. In the back, there will be a Mollier diagram for steam. Figure 24.2 is a representation of your Mollier diagram. We will use this diagram to calculate the amount of work we can recover from steam with a turbine:

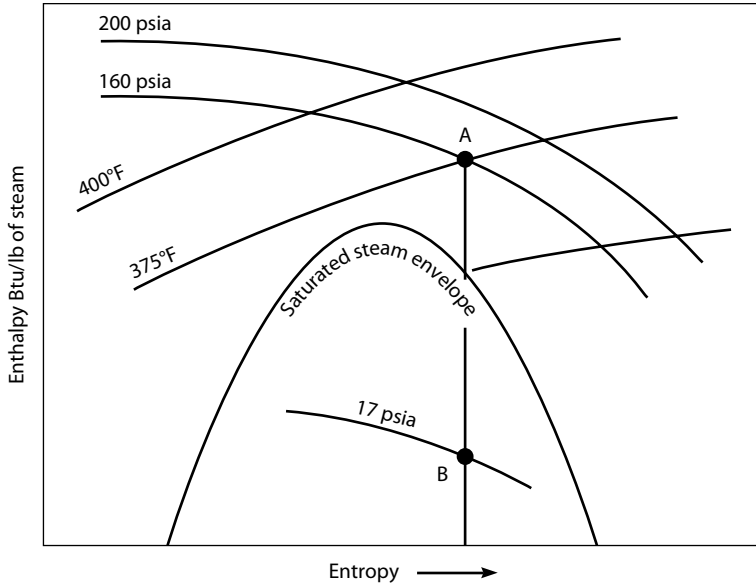


FIGURE 24.2 How to use a Mollier diagram.

1. The motive steam conditions are 375°F and 160 psia, which determine point A. Note that the motive-steam pressure is the pressure in the steam chest, not the supply steam pressure.
2. When steam passes through a turbine, it undergoes an isentropic expansion. The work that the steam does in transferring its momentum to the turbine wheel exactly equals the shaft horsepower developed by the turbine. The entropy of the system is therefore constant. On this basis, extend a line through point A straight down the Mollier diagram. This line represents a constant entropy expansion.
3. The exhaust-steam pressure is 17 psia. The intersection of the isentropic expansion line and 17-psia constant-pressure line together determine point B in Fig. 24.2.
4. Now measure the enthalpy difference in British Thermal Units per pound of steam between points A and B on the vertical (y axis) scale.
5. As we shall see in the following section, most of the increase in velocity or kinetic energy of the steam does not come from the pressure of the steam, but from its enthalpy. By enthalpy, I mean both the temperature and latent heat content of the steam. Thus, as steam expands through a steam nozzle to a

region of lower pressure in the turbine case, the steam cools and is partly condensed.

6. Divide your answer by 2500 Btu/hp. This is the fraction of a horsepower that each pound of steam can produce. However, as turbines are not 100 percent efficient, multiply the calculated horsepower by 0.90 to account for internal turbine inefficiencies called *windage losses*. Incidentally, the reciprocal of the calculated horsepower per pound of steam is called the *water rate*, which is the pounds of motive steam needed to generate 1 horsepower worth of shaft work.
7. Multiply the horsepower per pound of steam calculated in step 6 by the turbine steam flow in pounds per hour. This is the total shaft work that appears at the turbine's coupling. This is the amount of horsepower that is available to spin a centrifugal pump or a variable speed air blower.

24.1.3 Exhaust Steam Conditions

You may have noticed that point B in Fig. 24.2 is below the saturated-steam envelope line. Does this mean that the exhaust steam would have appreciable amounts of entrained water? Does this mean that the motive steam may sometimes partially condense inside a turbine? Answer—yes! Does this also mean that turbine exhaust steam may have too high a moisture content to be used in certain downstream services? Well, it is fine for reboilers, but it would be unsuitable for steam jets (see Chap. 23).

Looking at Fig. 24.2, see if you can agree with these statements:

- The higher the motive-steam superheat temperature, the drier the exhaust steam.
- The higher the exhaust-steam pressure, the drier the exhaust steam.

Lowering the exhaust-steam pressure always allows us to extract more work from each pound of steam. That is why we often exhaust steam to a condenser. But other than minimizing the exhaust-steam pressure, how else may we increase the amount of work that can be extracted from each pound of steam?

24.1.4 Horsepower Valves

It is the velocity of the steam impacting on the turbine wheel buckets that causes the turbine to spin. If that is so, then the way to extract more work from each pound of steam is to increase the velocity of the steam as it escapes from the steam nozzle, shown in Fig. 24.1.

The escape, or exit, velocity of this steam is a function of the steam pressure in the steam chest. If we raise the pressure in the steam chest by 30 percent, then the velocity of the steam leaving the nozzle would go up by 30 percent as well. It is true that I could simply open up the governor and reduce the pressure drop of the steam across the governor from 100 to 10 psi. This would raise the pressure in the steam chest from 300 to 390 psig. But I would also get 30 percent more pounds of steam flow through the nozzle. The turbine would spin a lot faster and develop a lot more horsepower. This is not my objective.

What I wish to achieve is to maintain the same horsepower output from the turbine. At the same time, I want to force open the governor speed-control valve, raise the pressure in the steam chest, but decrease the steam flow through the steam nozzle. The only way this can be done is to make the nozzle smaller.

We could shut down the turbine and unbolt the steam chest to expose the *nozzle block*, which is the wall that separates the steam chest from the turbine case. We could unscrew the existing nozzle and replace it with a smaller nozzle. A nozzle of 20 percent less diameter would reduce the nozzle cross-sectional area by 36 percent:

$$(1.00 - 0.20)^2 = 0.64 = 64\%$$

$$100\% - 64\% = 36\%$$

This procedure is called reducing the port size of the nozzle. It works fine, except that we have permanently derated the capacity of the turbine by 36 percent, and next month we might need this capacity.

24.1.5 Speed Valves

Figure 24.3 shows a steam turbine with three, rather than one, nozzles. The single, largest, left-hand nozzle is called the *main nozzle*. It handles 60 percent of the motive-steam flow. Each of the two smaller nozzles handles 20 percent of the steam. These 20 percent nozzles can be plugged off by a device sometimes called a *horsepower valve*, *jet valve*, *speed valve*, *star* (for the handle shape) valve, or *port valve*.

If we close off one of the two horsepower valves, steam flow into the turbine will drop—initially by 20 percent. The turbine will slow. This will cause the governor valve to open. The pressure drop across the governor will decrease. The pressure in the steam chest will rise. The flow of steam through the 60 percent port nozzle and the remaining 20 percent port nozzle will increase. The velocity of the steam striking the buckets will also increase. The turbine wheel will now come back up to its set point.

The net effect of this exercise will be to save not 20 percent of the motive steam, but 10 percent. The 20 percent reduction in nozzle area is partially offset by the opening of the governor valve. The inefficient, irreversible, isenthalpic expansion and pressure drop across the governor speed control valve is reduced. The efficient,

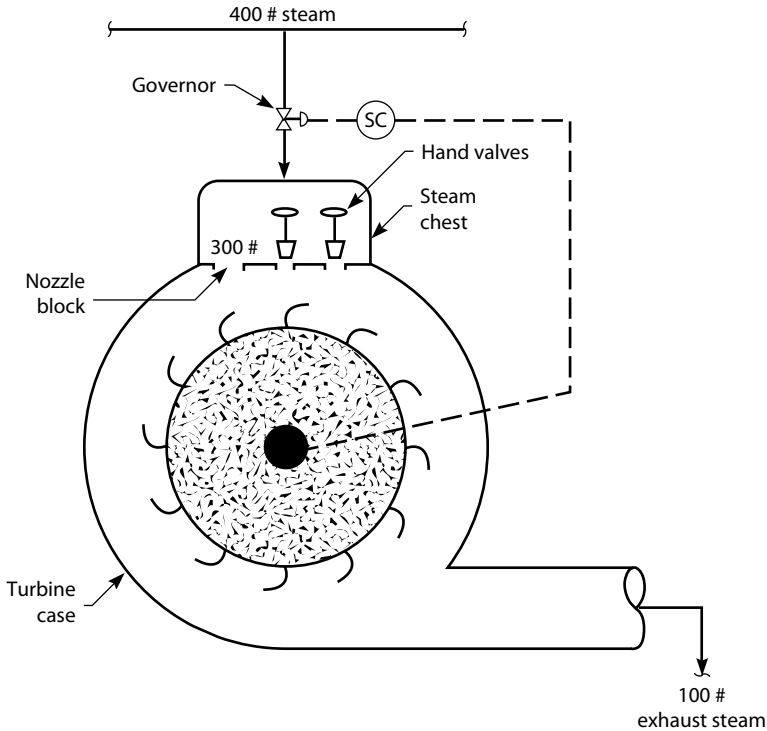


FIGURE 24.3 Use of hand valves can save steam.

reversible, isentropic expansion and pressure drop across the nozzles is increased.

If we now attempt to shut the second horsepower valve, we may be able to save a second 10 percent increment of steam. But if the governor goes wide open, we will have to open this horsepower valve again. You see, once the governor is 100 percent open, we will have lost our ability to control the turbine's speed. This is no good, because we want the turbine to run at a constant speed.

Do not pinch on the horsepower valves. They must be left completely opened or firmly closed. Leaving them in a partially open position will badly damage the valve seats by erosion, reduce the efficiency of the turbine, and make it impossible to close the valves in the future.

24.2 Selecting Optimum Turbine Speed

An ordinary American-type turbine is designed to run at 3600 rpm. Its overspeed trip will cut off the motive steam flow at about 3750 rpm. But the turbine can be run at any lower speed. There is usually a small knob on the left side of the governor-valve assembly that is used to

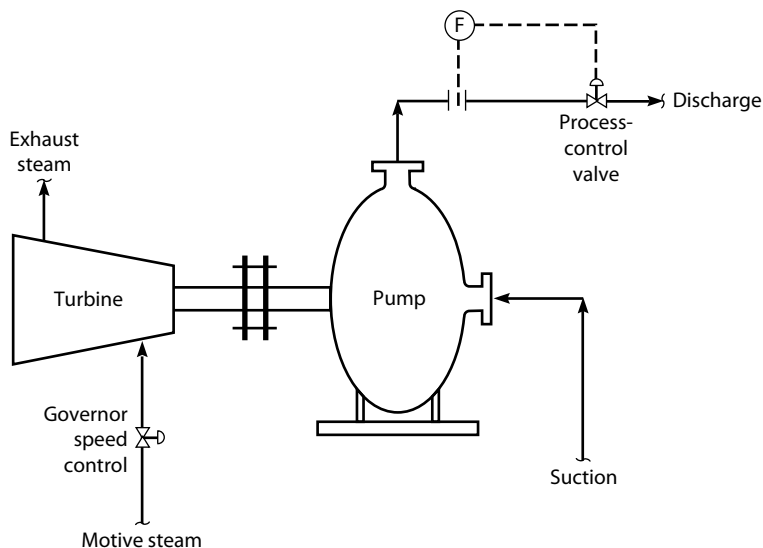


FIGURE 24.4 Selecting optimum turbine speed.

change the set speed. The question is, how do we know the best speed to operate the turbine?

Figure 24.4 shows a centrifugal pump driven by a steam turbine. The correct operating speed for the pump and turbine is the speed that puts the process-control valve in a mostly open, but still controllable position. As we slow the turbine to force open the process-control valve, the turbine's governor valve will close. Steam flow to the turbine will decline in accordance with *fan laws*:

$$\text{Shaft work} \sim (\text{speed of driver})^3$$

This means that if we drop the speed of the turbine by about 3 percent, motive-steam flow will decline by about 9 percent. However, there is an additional benefit.

Slowing a turbine closes the governor valve. This may now permit us to close an additional horsepower valve without losing our flexibility to control the speed of the turbine. Closing that final horsepower valve will save us another 10 percent of steam.

Our overall objective is to wind up with both the process-control valve downstream of the pump and the governor speed-control valve in a mostly wide open, but still controllable position. To achieve this dual objective, we have to simultaneously:

- Optimize the turbine speed
- Optimize the number of horsepower valves that are closed

It is rather like solving two equations with two unknowns.

24.2.1 An Old, But Better, Idea

The control scheme shown in Fig. 24.4 is certainly quite common. But is it the best? Figure 24.5 is a copy of the crude charge system in a now-defunct refinery in Port Arthur, Texas. I saw it in operation many years ago. It worked fine. The required flow of crude directly controls the governor. The turbine speed is then always at its optimum. The ΔP across the process-control valve is always zero, because there is no process-control valve. This design is a direct descendant of the original method of controlling the steam flow to pumps. The steam inlet valve was opened by the operator, so that the desired discharge flow was produced.

Why, then, do we need process-control valves on the discharge of variable-speed-driven pumps? Why, indeed!

24.2.2 Steam Rack

On larger steam turbines, we have automatically, rather than manually, operated horsepower valves. The mechanism that controls the movement of the horsepower valves is called a *steam rack*. If you have a large steam turbine driving a compressor, you have likely seen such a steam rack.

A horizontal rod extending across the width of the turbine has a half-dozen cams fixed along its length. These cams lift and lower, in sequence, plungers attached to the horsepower valves. Watch these plungers in action. If you slow the compressor, here is what will happen:

1. The governor valve will start to close.
2. The steam chest pressure will fall.

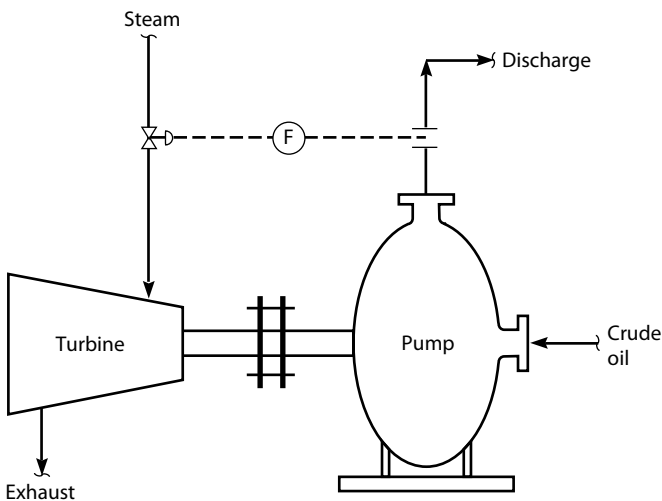


FIGURE 24.5 A better way to run a turbine.

3. The steam rack will move, so as to close only one of its multiple horsepower valves.
4. The steam chest pressure will rise, and the governor will open back up.

If you look on the local control panel of such a turbine, you will probably see four steam pressures displayed:

- Steam header pressure
- First-stage steam chest pressure
- Second-stage steam pressure
- Exhaust-steam pressure

These larger turbines often let down the steam in two or more stages. They are called *multistage turbines*. Their principle of operation is not significantly different from a simple single-stage machine.

24.2.3 Condensing Turbines

Most of the turbines you will encounter in your work are called “topping,” or *extraction*, turbines. The idea of such a turbine is to extract much of the potential work from the motive steam, and then use the exhaust steam to reboil towers. Typically, the energy content of the exhaust steam is only 10 to 20 percent less than that of the motive steam. That is the calculation we just did with the Mollier diagram. The rest of the energy of the steam may then be used as the steam condenses to reboil towers. This sounds pretty efficient. It is the basis for the new *cogeneration* projects you may have heard about. Of course, this system was used by the British Navy in the nineteenth century.

The *condensing turbine* does not produce exhaust steam. All the turbine exhaust steam is turned into water in a surface condenser. We will study surface condensers in Chap. 25. The surface condenser is just like the sort of vacuum condensers we discussed in Chap. 23. The exhaust steam condenses under a deep vacuum—typically 76 mm Hg, or 0.1 atm. Basically, then, a condensing steam turbine loses all the potential benefit of using its exhaust steam to reboil towers. But for what purpose?

Let’s say a turbine was using 400-psig (i.e., 415-psia) steam exhausting to a 30-psig (i.e., 45-psia) steam header. The work generated was 2000 hp. If we had condensed the steam at 76 mm Hg (i.e., 1.5 psia), the work generated would be 4000 hp. In the first case, the ΔP of the steam was 370 psi ($415 - 45 = 370$). In the second case, the ΔP of the steam is 413.5 psi ($415 - 1.5 = 413.5$).

If the ΔP increases by only 12 percent, why does the amount of work that is extracted from each pound of steam double? The answer is that it is not the pressure of the steam that drives the turbine. It is the velocity of the steam impacting on the turbine wheel buckets.

The velocity of the steam escaping from the steam nozzle (see Fig. 24.1) is 25 ft/s, when the exhaust pressure is 30 psig (45 psia). What, then, would be the velocity of the steam if the exhaust pressure were 76 mm Hg (1.5 psia)?

$$\frac{25 \text{ ft/s} \times 45 \text{ psia}}{1.5 \text{ psia}} = 750 \text{ ft/s}$$

No wonder the work we can extract from a condensing steam turbine is so much greater than with an extraction, or topping, turbine. The most efficient condensing turbines may convert 30 percent of the steam's energy into work. These turbines use 1500-psig steam, and exhaust to a surface condenser at 50 mm Hg. The most efficient machine ever built is the high-bypass, radial-flow jet turbine engine, commonly used in commercial aircraft, which has a 40 percent efficiency.

I rather like to adjust steam turbines. It reminds me of Professor Peterson, my thermodynamics instructor. It makes me think that the four years I spent at Cooper Union was time well spent. It is just nice to see how the Second Law of Thermodynamics functions in the real world.

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CHAPTER 25

Surface Condensers

The Condensing Steam Turbine

More of the world's energy is consumed in surface condensers than for any other single use. Even at the very start of the Industrial Revolution, the father of the surface condenser, the *barometric condenser*, consumed huge amounts of heat.

The central idea of our industrialized society is to have machines do the work formerly done by humans or animals. The technical problem that kicked off the Industrial Revolution was flooding in the tin mines of Cornwall, a region in southern Britain. To work the mines and extract the valuable tin ore, steam-driven reciprocating pumps had been constructed. These plunger-type pumps were moved up and down by the famous beam engines, and these reciprocating engines were powered by steam, thanks to the efforts of Thomas Newcomen in 1712.

The motive steam for these reciprocating engines was charged into a steam cylinder. The piston inside the cylinder was pushed up by the expanding steam. The piston then lifted the beam attached to the reciprocating pump. Cold water was admitted next and jetted into the steam cylinder. The cold water absorbed the latent-heat content of the steam. The steam pressure inside the cylinder dropped, and the piston dropped. This pulled the beam down. It was the up-and-down movement of the beam that powered the reciprocating pump. The structures, beams, foundations, and bits and pieces of ancient machinery are scattered all across Cornwall. (Caution: The tin mine entrances are rather obscure, overgrown with brambles, and your author more or less fell into one. If you choose to explore the mining areas of Cornwall, be very careful as many of these shafts are hundreds of feet deep. There are numerous stories about people and livestock slipping into these disused shafts and being drowned in the accumulated water or simply falling to their deaths down the mine shafts.)

There was a big problem with the initial beam engine design. When the cold water was admitted into the steam cylinder, the water not only absorbed the latent heat of the condensing steam but also cooled the iron walls of the steam cylinder. Then when the next

charge of steam was admitted to the cylinder from the boiler, a lot of the steam's heat was wasted in reheating the iron walls of the cylinder. Then again, an awful lot of cold water was wasted in cooling the metal of the steam cylinder each time the motive steam had to be condensed.

All this wasted a tremendous amount of coal. Actually, only 1 to 2 percent of the energy of the coal was converted into useful work. Quite suddenly, the steam engine was revolutionized. Its efficiency was increased by a factor of 10. This was all due to the innovations of James Watt, who invented the external barometric condenser in the late 1760s.

25.1 The Second Law of Thermodynamics

Steam power was fully developed before the introduction of the science of thermodynamics. The steam engine was designed and built by ordinary working people, such as Mr. Newcomen and Mr. Watt. Mr. Watt's invention is illustrated in Fig. 25.1.

Rather than cooling and condensing the steam in the cylinder, the steam was exhausted to an external condenser. In this external condenser, the exhaust steam was efficiently contacted with the cold water. This external or barometric condenser rather looks and performs like the deaerator we discussed in Chap. 21. The external condenser obviously achieved Mr. Watt's original objective. He could condense the steam without cooling the cylinder. But the barometric external condenser was found to have an even more important attribute. Let me explain.

The boilers in those days were not pressure vessels. They were constructed from sheets of wrought iron and assembled by riveting and hammering the seams. At best, they could hold a few atmospheres of steam pressure. The low-pressure steam generated did not really push the beam and piston up. The beam was pulled up by a heavy weight attached to the far end of the beam. This also pulled the piston up.

Well, if the steam did not really push the piston up, what did the steam do? Answer—the condensing steam pulled the piston back down. As the steam condensed, it created an area of very low pressure below the piston (see Fig. 25.1). This low pressure sucked the piston down. The colder the temperature at which the steam condensed, the lower the pressure at which the steam condensed.

The lower the pressure in the steam cylinder, the more forcefully the piston was drawn down. And the more forcefully the piston was sucked down, the more work the beam engine could do with the same amount of coal consumed.

It is easy to see how the barometric condenser could condense the exhaust steam more efficiently than periodically squirting water into the steam cylinder. The barometric condensers could absorb the latent heat of condensation of the steam at temperatures of 120°F or less.

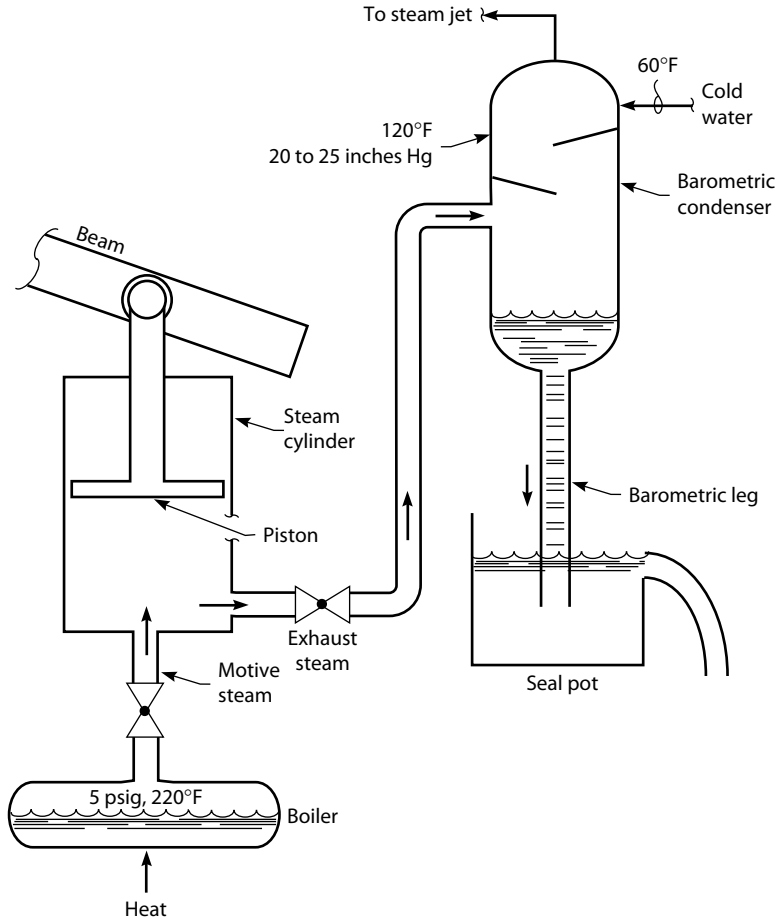


FIGURE 25.1 Barometric condenser; improved efficiency of the legendary beam steam engine.

Water condenses at a pressure of 25 inches Hg at this temperature. This extremely low vacuum sucked the piston down more forcefully because of the greater differential pressure across the cylinder. In other words, on top of the cylinder, there was atmospheric pressure; below the cylinder, there was the pressure in the barometric condenser.

Two problems arose with the use of the barometric condenser. First, if the condenser operated at sub-atmospheric pressure, how can the water be drained out of the condenser? That is easy. Set the condenser on a hill 34 ft high. Then drain the water down through a barometric leg to a seal pot. The pressure that a column of water 34 ft high exerts is equal to 1 atmosphere, or 1 bar. Hence the term barometric "leg."

The second problem was air leaks. Air drawn into the system would build up in the condenser. This non-condensable vapor was drawn off by using a steam jet.

Certainly, if we could generate steam at a higher pressure and temperature in the boiler, we could push the piston in Fig. 25.1 up with greater force. Thus thought Richard Trevithick of Cornwall, who pioneered the use of high-pressure steam in the 1790s. And as the mechanical design of boilers has improved over the last 200 years, this has been done, and we certainly can now push up the beam with greater force. As a result, the amount of work that may be extracted from steam has more than doubled. But it was James Watt, working alone, repairing a model of a Newcomen engine,¹ who made the big leap forward in improving the efficiency of the steam engine.

And what, dear reader, does all this have to do with the Second Law of Thermodynamics? This law states

$$W = (\Delta H) (T_2 - T_1) \quad (25.1)$$

where W = amount of work that can be extracted from the motive steam

ΔH = enthalpy of the motive steam minus the enthalpy of the condensed water

T_2 = temperature at which the steam is generated in the boiler

T_1 = temperature at which the steam is condensed in the barometric condenser

According, then, to the Second Law of Thermodynamics, Mr. Watt lowered T_1 . But, of course, the professors who worked out these laws were just formalizing the discoveries that practical working people had made 100 years before their time, using common sense and craftsmanship.

25.1.1 The Surface Condenser

There is another problem with the barometric condenser that did not become apparent at first. When the British Navy decided to convert from sail to steam, this problem was immediately obvious. While steam can be generated from seawater, it is far better to use freshwater, especially if one wishes to generate high-temperature, high-pressure steam. And as freshwater supplies are limited at sea, it would be great if the condensed steam could be recycled to the boilers. But the cooling-water supply to the barometric condensers was naturally seawater, which mixed with the steam condensate.

The solution is straightforward. Do not condense the steam by direct contact with cold water, as is done in the barometric condenser. Condense the steam by indirect contact with the cold surface of the tubes in a shell-and-tube condenser. Hence the name *surface condenser*, a sketch of which is shown in Fig. 25.2. Compare Fig. 25.1 with the

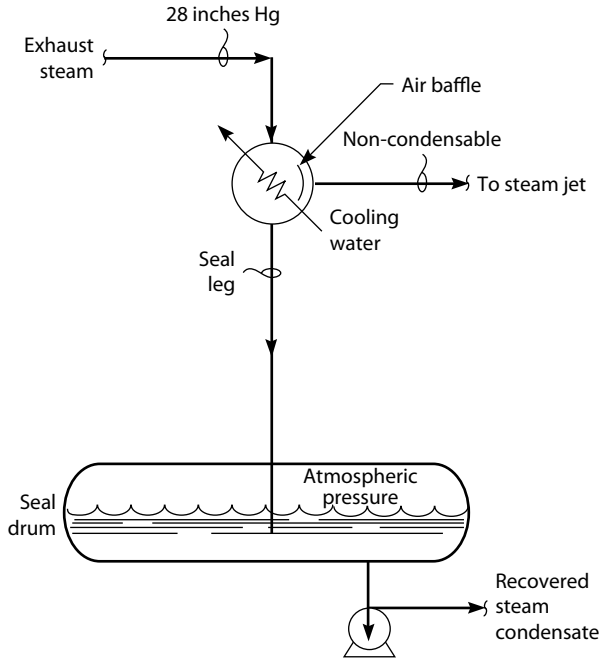


FIGURE 25.2 Surface condenser.

surface condenser. Is there really much difference? Other than recovering clean steam condensate for reuse, there is no difference at all. I last used a barometric condenser in 1976, on a sulfuric acid plant reactor feed gas boost blower, and it worked just fine.

25.1.2 Using the Second Law of Thermodynamics

The motive-steam supply to a condensing steam turbine such as that shown in Fig. 25.3 is 360°F and 150-psig saturated steam. The turbine is exhausting to a surface condenser. The cooling water to the condenser is 92°F. The turbine is driving a centrifugal compressor. The calculated horsepower produced by the turbine is 10,000 bhp (brake horsepower). Bill Duvall, your supervisor, has told you that colder 62°F well water is to be substituted for the 92°F cooling-tower water. Bill has given you the following additional information:

- The steam jets are oversized for the non-condensable flow, which consists of only a very few pounds of air in-leakage.
- The motive-steam flow to the turbine is not known, but won't change.
- The pressure in the surface condenser is unknown and cannot be measured.

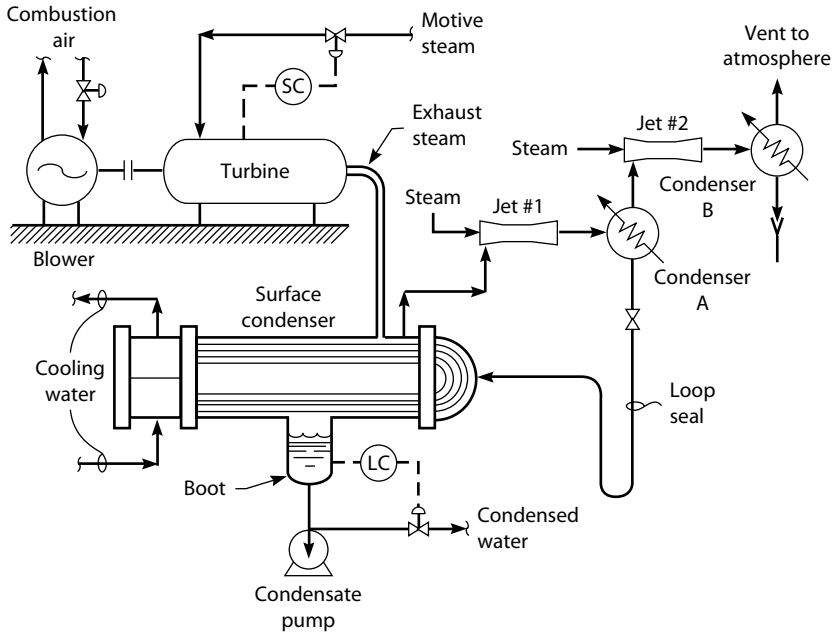


FIGURE 25.3 Condensing steam turbine driving an air compressor.

- The cooling-water flow rate is not known, but will not change either. The water will just be colder.
- The efficiency of the turbine and compressor is not known, but is presumed to remain constant.

Bill has asked you to calculate the new compression horsepower output from the compressor. Using Eq. (25.1), we note

- Compression work W is proportional to horsepower.
- ΔH will go up from its prior value a little because the enthalpy of the condensed steam will be lower (because it's colder).
- As the cooling-water supply is 30°F lower, we will assume that the condensation temperature T_1 in Eq. (25.1) is reduced by 30°F.
- The enthalpy difference between 150-psig saturated steam at 360°F and steam condensed under a good vacuum is roughly 1000 Btu/lb.
- The enthalpy reduction of condensing the steam at a 30°F lower temperature will increase ΔH by 30 Btu/lb.
- ΔH in Eq. (25.1) thus increases by 3 percent.
- T_2 , the temperature of the motive steam, is always 360°F. With 92°F cooling water, we will assume that T_1 is 120°F.

- $(T_2 - T_1)$, with 92°F cooling water, is $(360^\circ\text{F} - 120^\circ\text{F}) = 240^\circ\text{F}$.
- T_1 , with 62°F cooling water, is 30°F cooler than T_1 , with 92°F cooling water; that is, the new T_1 is $(120^\circ\text{F} - 30^\circ\text{F}) = 90^\circ\text{F}$.
- $(T_2 - T_1)$, with 62°F cooling water, is then $(360^\circ\text{F} - 90^\circ\text{F}) = 270^\circ\text{F}$.
- With the cooler well water substituted for the warmer cooling-tower water $(T_2 - T_1)$ has increased by

$$\frac{(270^\circ\text{F} - 240^\circ\text{F})}{(240^\circ\text{F})} = 12.5\%$$

Combining the ΔH effect of 3 percent with the larger $(T_2 - T_1)$ effect of 12.5 percent in Eq. (25.1) results in a compression horsepower increase of 16 percent or about 11,600 bhp total. If you wish, work through the same problem but use 5-psig saturated steam, which was used to power steam engines in the eighteenth century, rather than the 150-psig saturated steam we used in our current example. The answer will illustrate why Mr. Watt is still well remembered especially in Cornwall along with justly revered Mr. Trevithick.

Imagine you are sitting in a meeting when the vice president asks, "Should we use an aerial cooler or colder seawater to condense the power plant turbine exhaust steam?" Harry, the chief engineer, offers to run through the relative efficiency of the turbine for the two cases using his computer model and report back after lunch. You, the junior staff engineer, look up into space and after a few moments state, "Based on the Second Law of Thermodynamics I've calculated an enhanced efficiency of 16.1 percent for the cooler seawater case." I always add that extra decimal for dramatic effect.

25.2 Surface Condenser Problems

In Chap. 23 we reviewed several problems pertaining to steam jet pre-condenser and intercondenser problems. The surface condensers, which serve condensing steam turbines, are subject to all the same problems and a lot more. A standard surface condenser package with an associated two-stage jet system is shown in Fig. 25.3. By way of summarizing many of the problems that occur with this sort of equipment, I will relate my trials and tribulations with K-805, an auxiliary combustion air blower at the Good Hope Refinery in Norco, Louisiana (now Valero).

The problem with this new air blower was that we could not bring the turbine speed above its natural harmonic *critical speed*. The critical speed of a turbine is stamped on the manufacturer's nameplate. Turbines are typically run well above their critical speed. If, for some reason, a turbine is run close to its critical speed, it will experience uncontrolled vibrations and self-destruct.

For a surface condenser to work properly, non-condensable vapors must be sucked out of the shell side. This is done with a two-stage jet system, as shown in Fig. 25.3. When I first commissioned the jets, they were unable to pull a good vacuum. Moreover, water periodically blew out of the atmospheric vent. I found, after considerable investigation, that the condensate drain line from the final condenser was plugged.

I directed the maintenance crew to disassemble and clean the drain lines from both the final condenser (B) and the primary jet discharge condenser (A). Unfortunately, they failed to reassemble the *loop seal* from condenser A. But what is the purpose of this loop seal?

The pressure in condenser A is greater than that in the surface condenser and less than that in the final condenser (condenser B). This means that condenser A is operating at vacuum conditions. This prevents the condensed steam formed in condenser A from draining out to atmospheric pressure, unless the condenser is elevated by 10 to 15 ft. To avoid this problem, the condensate is drained back to the lower-pressure surface condenser. To prevent blowing the non-condensable vapors back to the surface condenser as well, a loop seal is required. The height of this loop seal must be greater than the difference in pressure (expressed in ft of water) between the surface condenser and the primary jet discharge condenser (condenser A).

With the loop seal gone, the non-condensable vapors simply circulate around and around through the primary jet, but no substantial vacuum in the surface condenser can be developed.

Having replaced the loop seal piping (some units use a steam trap instead of this loop seal), I started steam flow to the turbine. But the vacuum in the surface condenser, which had started out at an excellent 27 inches Hg, slipped down to 14 inches Hg. This loss in vacuum increased the back-pressure in the turbine case. The higher pressure in the turbine case reduced the velocity of the steam striking the buckets on the turbine wheel, which reduced the amount of work that could be extracted from each pound of steam.

Note that the vacuum measurements of 27 inches Hg and 14 inches Hg are expressed in the vacuum measurement system typically used in American refineries where:

- 30 inches Hg = zero mm Hg (i.e., full vacuum)
- 0 inches Hg = atmospheric pressure (i.e., no vacuum)

Thus 27 inch of Hg vacuum is really 3 inches of Hg absolute. And 14 inches of Hg vacuum is really 16 inches of Hg absolute. I know this is confusing, but that's the way vacuum pressure measurements and gauges are in process plants in the United States.

For practice, pull out your Mollier diagram. If the motive steam is 400 psig and 650°F, what is the effect of reducing the vacuum in the

surface condenser from 27 to 14 inches Hg? Answer: 13 percent loss in horsepower (see Chap. 24).

The turbine began to slow. It slowed to its critical speed and began to vibrate. Before shutting down the turbine to avoid damage due to the vibrations, I noted the following:

- The temperature of the turbine vapor exhaust to the primary jet had increased from 125 to 175°F.
- The temperature of the condensate draining from the surface condenser had decreased from 125 to 100°F.
- The condensate pump was cavitating, as indicated by an erratically low discharge pressure.

The increase in the vapor outlet temperature from a condenser, as compared to a decrease in the temperature of the condensate from the same condenser, is a sure sign of condensate backup. The condensate is covering some of the tubes in the surface condenser. This subcools the condensate and does no harm.

However, the number of tubes exposed to the condensing steam is also reduced. This forces the steam to condense at a higher temperature (as discussed in Chaps. 12 and 18). In effect, the condensate backup has reduced the surface area of the condenser available to condense the steam. The higher the condensation temperature of the steam, the higher the condensation pressure of the steam. Just like the deaerator I described in Chap. 21.

Take a look at Fig. 25.3. It is the vapor outlet temperature of the surface condenser, rather than the condensate outlet temperature of the surface condenser, that determines the real condensing temperature and pressure of the exhaust steam.

Condensate pumps serving surface condensers have a common problem. Their suction is under a vacuum. For example, let's assume the following for Fig. 25.3:

- The surface condenser pressure = 27 inches Hg.
- The condensate water level in the boot is 11 ft above the suction of the pump.
- One inch of mercury (1 inch Hg) is equal to a head of water of 1.1 ft (this is a good rule of thumb worth remembering).

The pressure at the suction of the condensate pump is then

$$27 \text{ inches Hg} - 11 \text{ ft H}_2\text{O} \times \frac{1 \text{ inch Hg}}{1.1 \text{ ft H}_2\text{O}} = 17 \text{ inches Hg}$$

Often, centrifugal pumps develop seal leaks. If the suction of a centrifugal pump is under a vacuum, air will be drawn into the pump

through the leaking seal. The pump's capacity will be severely reduced. To stop the suspected air leak, I sprayed water from a hose over the pump's seal. Now, instead of sucking air, the leaking seal drew in cold water. As a result

1. The cavitation of the condensate pump stopped.
2. The high water level in the boot was pulled down.
3. The condensate outlet temperature increased.
4. The vapor (or non-condensable) outlet temperature decreased.
5. The vacuum in the surface condenser was restored.
6. And the turbine speed came back up, well above its critical speed.

But not for long. After 15 minutes of operation, the turbine speed slipped back down. Once again, I had lost a lot of vacuum in the surface condenser. Once again, the vapor outlet temperature had dramatically increased. But this time the condensate outlet temperature had also increased. What was my new problem?

I now observed that the surface condenser cooling-water outlet temperature had increased from 100 to 135°F. This is a sign of loss of cooling-water flow. As none of the other water coolers in the plant had been affected, I concluded that the cooling-water inlet to my surface condenser was partly plugged.

I had the front endplate on the cooling-water side of the surface condenser (called the *channel head cover*) removed. Most of the tube inlets in the *channel head tubesheet* were plugged with crayfish (but in Louisiana, where this story is set, everyone calls these little creatures crawfish).

The offending wildlife were removed. The condenser was reassembled. The motive steam was started to the turbine. Both the turbine and the air blower were running well above critical speed. We lined the flow of combustion air up to the combustion chamber. Everything was finally going my way, except for one minor problem.

The governor speed-control valve, shown in Fig. 25.3, was 100 percent open. The plant's boiler house was not sending us the proper pressure steam. I was supposed to be getting 460-psig steam, but was only receiving 400-psig steam. This reduced the steam flow through the nozzles in the turbine's steam chest by 15 percent. The operators at the power station assured me that the problem was temporary. The normal 460-psig steam pressure would be restored by morning.

Ladies and gentlemen, it is not a good idea to run a turbine with the governor speed-control valve wide open. Why? Because you no longer have any speed control. And the turbine speed is then free to wander. The rest of this story is pure philosophy.

During the evening, my operators decided to increase the combustion air flow from the blower or air compressor. This is done by

opening the suction valve to the blower. Naturally, it requires more work to compress more air. But the turbine could not produce any more work or horsepower because the steam turbine's governor speed-control valve was already 100 percent open!

So the turbine slowed down. And what was the only possible speed that it could slow down to? Why, the critical speed, of course. The turbine and blower began to vibrate. The bearings were damaged. The turbine's rotor became unbalanced. A rotating element on the air blower touched a stationary component in the blower's case. The stationary component broke off and wrecked the blower. That was the end of K-805.

By what law of nature was the turbine forced to slow exactly to its critical speed? You see, dear reader, life is perverse. And if anything bad can happen, it is going to happen to me.

25.2.1 Non-condensable Load

The gas that accumulates inside the surface condenser is called the *non-condensable load to the steam jets*. Some of the non-condensable load consists of CO_2 accidentally produced when the boiler feedwater is vaporized into steam. Air leaks through piping flanges and valves are other sources of non-condensable vapors. But the largest source of non-condensable vapors is often air drawn into the turbine case through the shaft's mechanical seals. To minimize this source of leaks, 2 or 3 psig of steam pressure is ordinarily maintained around the seals. However, as the turbine's shaft seals deteriorate, air in-leakage problems can overwhelm the jet capacity. This will cause a loss of vacuum in the surface condenser.

If vacuum in the surface condenser is bad, there are two possible causes. Either the jets are at fault or the surface condenser is at fault.

Air leaking into the surface condenser results in *vapor binding*. This causes an increase in the shell-side heat transfer thermal resistance. In effect, the steam does not make good contact with the exterior of the surface condenser tubes. This vapor-binding effect often appears to the operators as if the tubes are fouled with cooling-water deposits.

Thus, even without any loss in vacuum due to an air leak, the performance of a surface condenser is degraded as a consequence of an increasing concentration of air or non-condensables inside the surface condenser.²

Whereas shell-side fouling in a surface condenser is uncommon, loss of heat transfer coefficient due to air binding is as common as tube-side fouling due to poor-quality cooling water.

25.2.1.1 Jet Problems

These include too high or too low motive-steam pressure, excess wear on the steam nozzles, high condenser back-pressure, and air leaks that exceed the jet's capacity. To determine whether a poor vacuum in a surface condenser is due to such jet problems, consult the

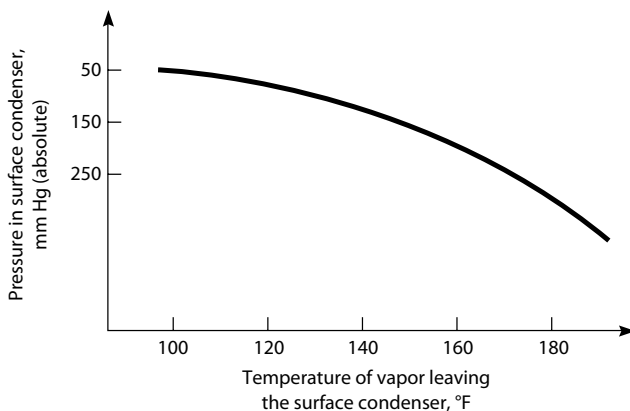


FIGURE 25.4 Vapor pressure of water under vacuum.

chart shown in Fig. 25.4. Measure the surface condenser vapor outlet temperature and pressure. Plot the point on the chart. If this point is somewhat below the curve, your surface condenser's loss of vacuum is due, at least in part, to jet deficiencies.²

25.2.1.2 Surface Condenser Problems

These include undersized surface condenser area, water-side fouling, lack of water flow, condensate backup, leaking seal strips around the air baffle, and excessive cooling-water inlet temperature. To determine whether a poor vacuum in a surface condenser is due to such heat-transfer problems, plot the surface condenser vapor outlet temperature versus pressure on the chart shown in Fig. 25.4. If this point is on or slightly below the curve, it is poor heat transfer in the surface condenser itself that is hurting the vacuum.

The curve in Fig. 25.4 also represents the best possible vacuum that can be obtained in any surface condenser. The majority of surface condensers I have seen do operate right on the curve. Condensers operating below the curve are typically suffering from air in-leakage through the turbine shaft seals.

25.2.1.3 Refinery Vacuum Precondenser Fouling

A precondenser in a refinery vacuum tower service is like any surface condenser, with the following characteristics:

- Triangular tube pitch
- $\frac{3}{4}$ -inch tubes
- 15/16-inch tube spacing
- Fixed tubesheet

The fixed tubesheet configuration means that the bundle cannot be extracted from the shell. The small tube pitch configuration makes

it utterly impossible to chemically clean the shell side in place for any shell ID much above 30 inch.

Recently, we had a refinery precondenser that had a 25 mm Hg ΔP . We tried the following chemical cleaning procedure:

- Step 1—Steam through with an aggressive vapor-phase solvent used to clean tower internals.
- Step 2—Circulate methanol for several hours and flush with water.
- Step 3—Circulate toluene for several hours and flush with water.
- Step 4—Circulate HCl for several hours and flush with water.
- Step 5—Steam through.

Afterward, ΔP had dropped from 25 to 20 mm Hg (calculated clean ΔP was 5 mm Hg). Cost of the cleaning was \$130,000. My client was not pleased with this result.

The correct design, then, for this service is:

- Pull-through bundle that can be extracted from the shell for cleaning
- One-inch O.D. tubes
- Rotated square pitch
- 1½-inches tube spacing

A ¾-inch tube on 1¼-inch spacing, I would imagine, is also cleanable.

A fixed tubesheet exchanger in a refinery precondenser service for a vacuum tower is a throwaway item. One of my clients just engaged in this exercise. Replacement for the 60-inches ID shell, as per my specs above, was \$1,300,000 (in 2013). Not a happy situation.

25.2.2 Function of the Final Condenser

We discussed before that the drain from the final condenser shown in Fig. 25.3 had plugged. Rather than unplugging the drain, could we have simply disconnected the final condenser (condenser B) and vented the discharge from the secondary jet (jet 2) to the atmosphere? Would this have helped or hurt the vacuum in the surface condenser?

The final condenser hurts the upstream vacuum. The final condenser increases the discharge pressure from the secondary jet and thus makes the jet system work slightly harder to expel the non-condensable gas load.

What, then, is the true function of the final condenser? Well, if the tiny amount of condensed steam is not needed, the final condenser serves no function at all. It may safely be discarded. Why, then, do surface condensers come with final condensers? It is just a convention that, for most plants, makes no particular sense. It is really just a

holdover from the design to conserve freshwater on the old British naval steamships.

25.2.3 Leaking Ejector Condenser Partition Plate

On the vapor outlet of steam turbine surface condensers, there is a two-stage ejector system and a two-stage condenser. Referring to Fig. 25.5, this two-stage condenser is actually a small condenser divided into two portions on the shell side by a partition baffle.

What will be the consequences of a leak developing in this partition baffle? I just encountered this problem last week for the first time, although I now understand that it is not uncommon.

For clarity, let's assume for Fig. 25.5 that valve A is closed and valves B and C are open. Note that the inlet pressure to the second-stage jet (380 mm Hg) is double its design pressure. Hence the poor vacuum (100 mm Hg) in the upstream surface condenser. If a leak develops in the partition plate in the ejector condensers, then non-condensables (i.e., air) will leak back from the 780 mm Hg discharge

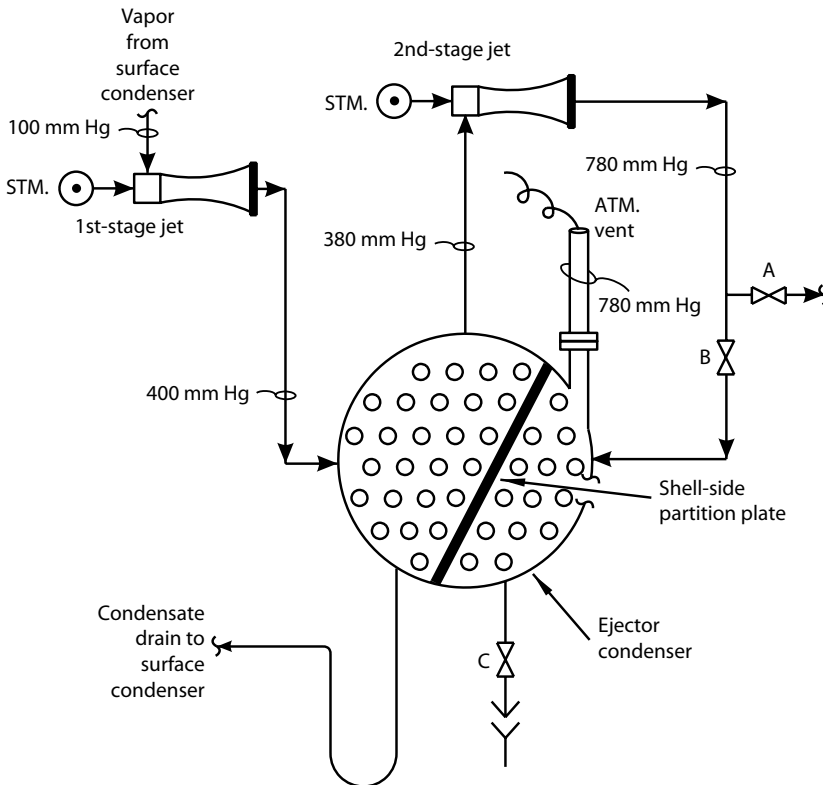


FIGURE 25.5 Steam turbine surface condenser vacuum is bad due to a leak in the ejector condenser shell-side partition plate.

of the second-stage jet into the 380 mm Hg suction of the second-stage jet. This internal non-condensable leakage will cause internal recirculation and overloading of the second-stage jet. Hence the poor performance (i.e., only a $780 \div 380 = 2.05$ compression ratio) of the second-stage jet.

The symptoms of this problem were:

- Closing valve C improved the surface condenser vacuum by 30 mm Hg.
- Closing valve C did *not* cause water to blow out of the atmospheric vent.
- With valve C open, the atmospheric vent was under a slight negative pressure.
- With valve C shut, the atmospheric vent began to blow out non-condensables (air) in an erratic manner.

The solution to this problem was not to repair the defective ejector condenser. This had been done before, but without preventing a repeat failure. As I've explained in Section 25.2.2 of this text, the final or second-stage condenser serves no function. It's a design error. So in this case, as shown in Fig. 25.5, valves B and C were shut and the local atmospheric vent valve A was opened. This allowed the second-stage jet to discharge directly to the atmosphere. As a result, the pressure in the surface condenser dropped by about one-half.

In reality, valve A did not exist. The second-stage jet was vented to the atmosphere by disconnecting the piping at the outlet flange of the ejector.

25.3 Surface Condenser Heat-Transfer Coefficients

Heat-transfer coefficients in this book have the units of $\text{Btu}/[(\text{h})(\text{ft}^2)(^\circ\text{F})]$, where the ft^2 term refers to the surface area of the surface condenser. The $^\circ\text{F}$ term refers to the condensing steam temperature minus the average tube-side cooling-water temperature.

Most unfortunately, an incorrect correlation for heat-transfer coefficients for surface condensers has become widely disseminated in several books devoted to heat transfer. This correlation predicts heat-transfer coefficients for clean condensers of about 650, when the water-side velocity is about 6 ft/s. Use of this correlation has led to some extremely serious problems, with which your author is intimately acquainted.

The correct heat-transfer coefficient for a clean surface condenser, with a water-side velocity of 6 ft/s, is about 200 to 240. Including an allowance for fouling, we suggest you use 140 to 160 overall heat-transfer coefficient for steam surface condensers. While I have

observed clean coefficients approaching 400, I would not count on maintaining 400 after several years of service on an industrial surface condenser.

25.3.1 Effect of Air on Condensate Film Heat Transfer

I've been helping a young engineer working at a plant in Norco, Louisiana, with a steam surface condenser problem. Actually, this young engineer is very smart. Smart enough not to follow my advice. For over a year I've been telling him that the cause of his poor surface condenser performance was a low heat-transfer coefficient due to fouling on the condenser shell side. This turned out to be 100 percent correct.

The poor heat transfer in the shell side of the surface condenser was increasing condenser vapor outlet temperatures, and hence the vapor pressure of water in the condenser outlet. Also correct.

The high water vapor pressure was thus limiting vacuum. Correct.

Therefore, I concluded that the shell side of the condenser must be fouled with hardness deposits from the poor-quality motive steam to the turbine. Thus, the shell side needs to be chemically cleaned. One hundred percent wrong!

Recently, the condenser heat-transfer coefficient was doubled just by improving the performance of the first-stage ejector. This was done by improving condensate drainage from the lead interstage condenser on the discharge of the first-stage ejector. As the amount of non-condensables (i.e., air) plus steam flow from the surface condenser was increased, the interface resistance between the cooling water tubes and the condensing steam was greatly reduced. This doubled the observed overall heat-transfer coefficient in the surface condenser. A vast increase (5 inches Hg or 126 mm Hg) in condenser performance was achieved.

This has nothing to do with the effect on the partial pressure of water due to air leaks. It is really "fouling" of the exterior of the tubes with air. I know this is complex, but it is all explained and quantified in the literature. This problem is best described by the term vapor binding.

To restore drainage from the lead interstage condenser, the engineer flushed out the drain line, which was full of black gunk. The literature reference is Henderson and Marchello (1969), "Film Condensation in the Presence of Non-Condensibles," *Journal of Heat Transfer*, Vol. 91, August, pp. 447–450. Also, I have a message for Misters Henderson and Marchello: "God sees the truth, but waits." As for the young engineer, the hero of the story, he's still waiting for his reward from the owners of that plant in Norco.

References

1. B. Trinder, ed., *The Blackwell Encyclopedia of Industrial Archaeology*, Blackwell, London, 1992.
2. Richard E. Putman, *Steam Surface Condensers*, The American Society of Mechanical Engineers, ASME Press, New York, 2001, pp. 136–137.

CHAPTER 26

Shell-and-Tube Heat Exchangers: Heat-Transfer Fouling Resistance

After distillation, heat transfer is the most important operation in a process plant. Most of the heat transfer in chemical plants and petroleum refineries takes place in shell-and-tube heat exchangers. The surface condenser we discussed in Chap. 25 is an example of a shell-and-tube heat exchanger.

A wide variety of heat exchangers are available, some of which you may have seen: plate, spiral, and coil—to name just three. But 99 percent of the heat exchangers I have worked with are ordinary shell-and-tube exchangers, the design of which has not changed since the 1920s.

Before considering the process aspects of heat transfer, let's look at the mechanical components of the heat exchanger shown in Fig. 26.1. This shell-and-tube heat exchanger is actually a compromise between four aspects of heat-exchanger design:

- Allowance for thermal expansion
- Efficient heat transfer
- Ease of cleaning
- Mechanical robustness

26.1 Allowing for Thermal Expansion

Referring to Fig. 26.1, we can see how a *floating-head exchanger* works. The tube-side flow enters the bottom of the *channel head*. This assumes the cold fluid to be on the tube side. The cold fluid may be on the shell side or the tube side of an exchanger. The convention is to put the

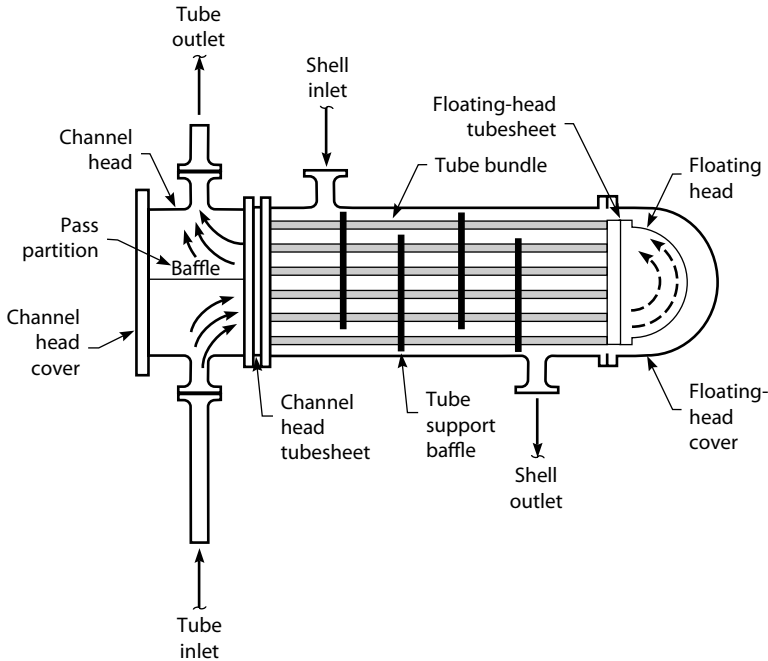


FIGURE 26.1 Shell-and-tube heat exchanger. Tube support baffles 90° from true orientation.

cold fluid nozzle on the bottom of the exchanger. Sometimes this is necessary. Sometimes it does not matter, but it is still the convention.

Inside the exchanger's channel head, we have the *pass partition baffle*, which divides the channel head into two equal portions. This baffle forces the total flow only through the bottom half of the tubes. The tubes themselves are pipes of either 0.75 or 1 inch OD (outside diameter). The front end of each tube is slipped into a slightly larger hole drilled into the *channel head tubesheet*. This tubesheet is a disk about 2 inch thick, slightly larger than the inner diameter of the shell (shell ID).

The tubes are firmly attached to the tubesheet by "rolling." After a tube is pushed into the tubesheet, a tapered tool is inserted into the open end of the tube and forcefully rotated. The tube's diameter is thus slightly expanded. While rolling is quite effective in sealing the tube inside the tubesheet, rolls have been known to leak.

The tube-side fluid now flows into the *floating head*, which acts as a return header for the tubes. The tube-side flow makes a 180° turn and flows back through the top half of the *floating-head tubesheet*. The floating head is firmly attached to the floating-head tubesheet. But why is it that one end of the tubes must be left free to float? The reason is thermal expansion—or, more precisely, the differential rate of thermal expansion between the tubes and the shell.

Not all shell and tube exchangers have a floating head. Many exchangers have individual U bends for each tube. In such cases, each of the U bends functions like a mini-floating head for each tube.

The requirement to leave one end of the tubes free to float creates a rather unpleasant process problem. The most efficient way to transfer heat between two fluids is to have true countercurrent flow. For a shell-and-tube exchanger, this means that the shell-side fluid and the tube-side fluid must flow through the exchanger in opposite directions. When calculating the *log mean temperature driving force* (LMTD), an engineer assumes true countercurrent flow between the hot fluid and the cold fluid.

But this is not the case with a floating-head exchanger. The tube-side fluid reverses direction in the floating head. It has to. There is no way to attach the tube-side outlet nozzle to the floating head. It is a mechanical impossibility. So we bring the tube-side fluid back to the top half of the channel head. Half of the tubes are in countercurrent flow with the shell-side flow. And that is good. But the other half of the tubes are in concurrent flow with the shell-side flow. And that is bad.

When calculating the LMTD for such a floating-head exchanger, the engineer has to apply the *F*-factor derating coefficient to reflect the loss in heat-transfer efficiency because of the floating head. A typical exchanger might lose 5 to 30 percent of its capacity because of non-true countercurrent flow. (See D. Kern, "Heat Transfer Fundamentals," for calculation of *F*-factors.)

26.1.1 Designing to Allow for Thermal Expansion

26.1.1.1 One-Pass Tube-Side Exchangers

I have not been completely accurate in the preceding discussion. Many exchangers are designed to bring the tube-side fluid in one end of the shell and out the other side. This is a true countercurrent arrangement. To provide for thermal expansion in such exchangers, an *expansion joint* is provided. These expansion joints are prone to leaking. They represent inferior mechanical engineering practice.

For moderate-temperature and low-pressure condensers, such as vacuum surface condensers (see Chap. 25), a single pass on the tube side is not uncommon. These exchangers are typically fixed-tubesheet designs. Such exchangers are fine from a purely process point of view. However, there is no practical way to disassemble the exchanger to clean the outside (i.e., the shell side) of the tubes. This inability to clean the shell side frequently leads to tremendous loss of efficiency after the condensers foul.

26.1.1.2 Shell-Side Flow

The hot shell-side flow enters the exchanger, as shown in Fig. 26.1, through the top inlet nozzle. Not shown on this sketch is the *impingement plate*, which is simply a square piece of metal somewhat larger

than the inlet nozzle. Its function is to protect the tubes from the erosive velocity of the shell-side feed. The plate lies across the upper row of tubes.

The four *tube support baffles* shown in this exchanger (see Fig. 26.1) serve a dual function:

- They serve to support the tubes.
- More importantly, they promote high *cross-flow velocity*.

The concept of cross-flow velocity is quite important in understanding how heat exchangers work. This concept is related to a flow phenomenon called *vortex shedding*. Perhaps you have seen a wire quivering in the wind. What causes the wire to vibrate with such energy? What causes water to swirl around a tree stump in a river?

When a fluid such as air or water flows perpendicularly across a wire or tube, vortices, such as those shown in Fig. 26.2, are created. The resulting turbulence that forms behind a wire will cause the wire to vibrate. The turbulence that forms behind a tube in a heat exchanger promotes good heat transfer.

Turbulence always encourages good sensible-heat transfer. The greater the velocity, the more violent the turbulence. For example, I recall a -40°F day in Fort McMurray, Alberta. The air is quite still, and I am comfortable. Suddenly the wind comes up. The rate of sensible-heat transfer from my body to the ambient air increases tenfold. Why does this happen?

Our bodies are always surrounded by a thin layer or film of stagnant air. This film is like a layer of insulation. It retards heat transfer between our skin and any surrounding fluid. Movement of the fluid

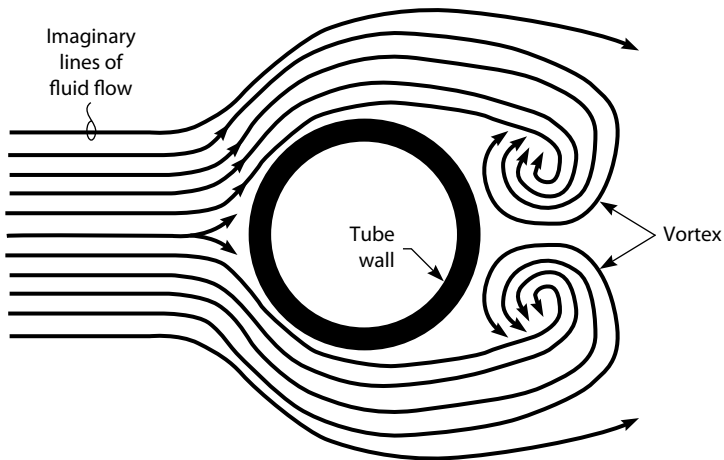


FIGURE 26.2 Liquid flow across a tube creates vortices and turbulence, thus improving heat transfer.

causes turbulence. The turbulence disturbs the film and reduces the film's resistance to heat transfer.

To encourage vortex shedding and turbulence on the shell side of a heat exchanger, we must increase the cross-flow velocity. To calculate the cross-flow velocity, we proceed as follows:

1. Find the distance between the adjacent tube support baffles, in inches.
2. Count the number of tubes at the edge of the tube support baffle. As shown in Fig. 26.3, this would be 7.
3. Multiply the number of tubes by the space between each tube, in inches.
4. Multiply the inches measured in step 1 by the inches calculated in step 3.
5. Divide by 144 to obtain the shell-side cross-flow area in square feet.
6. Divide the pounds per second of shell-side flow by the fluid density in pounds per cubic foot. This will give you the volumetric flow in ft^3/s .
7. Divide the volume calculated in step 6 by the area calculated in step 5. This is the cross-flow velocity.

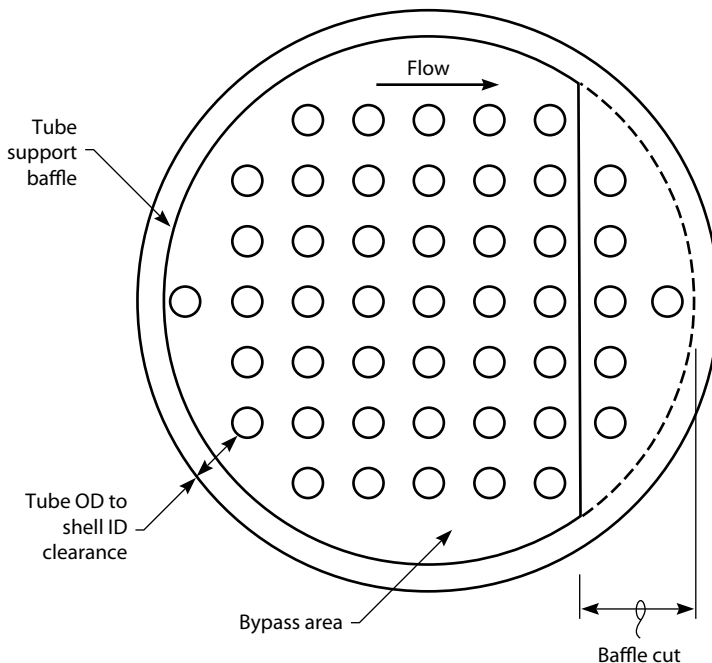


FIGURE 26.3 Tube support baffle.

A good cross-flow velocity for water is 3 to 5 ft/s. For fluids other than water, a reasonable cross-flow velocity, in feet per second, is

$$\frac{30}{(\text{Density})^{1/2}}$$

where the *density* term is the density of the shell-side fluid, in pounds per cubic foot (lb/ft³).

The baffle cut, shown in Fig. 26.3, is usually about 20 to 30 percent of the diameter of the baffle. The smaller the baffle cut, the more perpendicular the flow across the tubes. Perpendicular flow encourages desirable cross-flow velocity and vortex shedding. But a smaller baffle cut will also increase the pressure drop on the shell side.

26.1.1.3 Effect of Shell-Side Pressure Drop

Reducing the baffle spacing increases cross-flow velocity and improves heat transfer. But it also increases the shell-side pressure drop. Reducing the baffle cut also improves heat transfer, but increases ΔP .

Too much shell-side pressure drop can create a problem. The problem is flow through the bypass area shown in Fig. 26.3. This bypass area is caused by three factors:

- The tube support baffles must have a diameter somewhat smaller than the ID of the shell.
- The holes drilled in the baffles for the tubes cannot be drilled too close to the edge of the baffles.
- The use of an impingement plate requires the removal of several rows of tubes. This creates a large gap between the edge of the tube bundle and the shell. It is the primary purpose of the seal strips to reduce this gap (discussed in the next section). For the seal strips to fulfill this function, one of the seal strips in each pair must extend the full length of the tube bundle. This is rather complicated to explain. However, if you will look at a tube bundle that has been extracted from its shell, you will understand why this is needed, and how impingement plates render seal strips essential for good sensible heat transfer from a shell-side fluid.

The gap thus created between the shell ID and the outer row of tubes will permit the shell-side fluid to bypass around the tubes. This is obviously very bad for heat transfer. And as the shell-side ΔP increases, the percent of fluid that is squeezed through the bypass area increases.

If the baffle spacing gets too small, the shell-side heat-transfer rate will actually worsen. This happens even though the cross-flow velocity increases. What can be done to correct this problem?

26.1.1.4 Shell-Side Seal Strips

See if you cannot find an old tube bundle lying around your plant. Many such bundles have pairs of metal strips set around the edge of the tube bundle. These metal strips are typically 0.25 inch thick and 4 inch wide. They extend down the length of the tubes. As seen in Fig. 26.4, they are inserted in grooves cut in the tube support baffles. These seal strips often increase heat-transfer efficiency by 5 to 10 percent.

The function of seal strips is to interfere with, and hence reduce, the fluid flow through the bypass area. Often, one pair of seal strips is used for every 18 inch of shell ID (inner diameter). These seal strips encourage good shell-side cross-flow velocity and also help reduce localized fouling caused by low velocity.

Please note that the tube support baffles are normally installed with the baffle cut in a vertical position. I have shown the baffle cut in Fig. 26.1 in a horizontal position for clarity. Baffles are installed vertically in most exchangers to reduce buildup of sludges in the bottom of the shell and avoid trapping vapors in the top of the shell.

For exchangers with vertically cut baffles, the shell-side impingement plate will be opposite the shell-side inlet nozzle. Contrary to the location of the seal strips shown in Fig. 26.4, the critical seal strips will be set on either side of the impingement plate (see Fig. 28.1 in Chap. 28). Additional pairs of seal strips can also be installed as shown in Fig. 26.4. (See Chap. 28, "Shell-and-Tube Heat Exchangers:

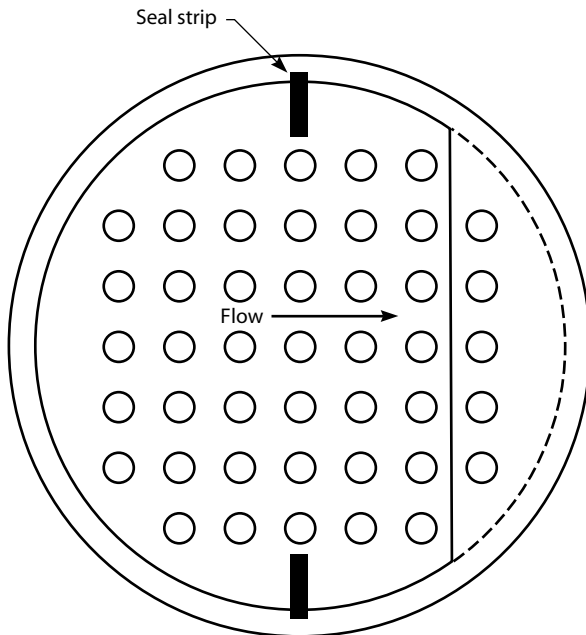


FIGURE 26.4 Seal strips reduce bypassing around tube bundle.

Design Details,” for more information on the use of these critical seal strips.)

26.1.1.5 Terminal-Tube Velocity

Have you ever seen a heat-exchanger tube bundle pulled? Perhaps many of the tubes were bent and twisted like partly cooked spaghetti. This distortion could not be very good for the shell-side heat transfer. The cause of the bent tubes is called *terminal-tube velocity*. Let me explain.

Let’s assume we have a new heat exchanger. The fluid on the tube side is crude oil. Crude oil, like many process fluids, will eventually foul and plug heat-exchanger tubes. The tendency to lay down fouling deposits is accelerated by

- Low velocity
- High temperature
- Dirt, salts, corrosion products, and other particulates

The crude oil is being heated by a hot-oil stream circulating through the exchanger’s shell. Even though the exchanger is new, the flow of crude through all the tubes is not equal. Some of the tubes naturally run somewhat slower, and in some of the tubes, heating the crude oil will run faster.

Those tubes that start running slower tend to get hotter. As the crude oil in these tubes gets hotter, the growth of fouling deposits accelerates. As the buildup of deposits in the slower tubes increases, the flow through these tubes is further restricted. The lower velocity increases fouling rates and restricts flow. This also increases the temperature of the slower tubes and accelerates the rate of fouling. These tubes continue to plug and foul, and run slower, until crude-oil flow is essentially lost.

Meanwhile, the flow of crude oil through those tubes that started out running faster increases. The crude-oil flow is backed out of the slower tubes by the fouling deposits. The extra flow increases the velocity in the tubes that initially ran faster. The higher velocities retard the buildup of fouling deposits. Also, the greater flow keeps these tubes cooler. This also discourages the accumulation of deposits inside the tubes.

To summarize, some of the tubes in the bundle continue to foul until they plug off. These tubes get hotter until they reach the temperature of the hot oil circulating through the shell. Other tubes continue to receive more and more flow. The velocity in these tubes increases to the point where the rate of fouling becomes inconsequential. This velocity is called the *terminal-tube velocity*. These tubes may run 50 to 150°F cooler than the plugged tubes.

26.1.2 Differential Rates of Thermal Expansion

It is quite true that the floating head permits differential rates of thermal expansion between the shell-and-tube bundle of an exchanger.

However, the floating head cannot permit differential rates of thermal expansion between individual tubes.

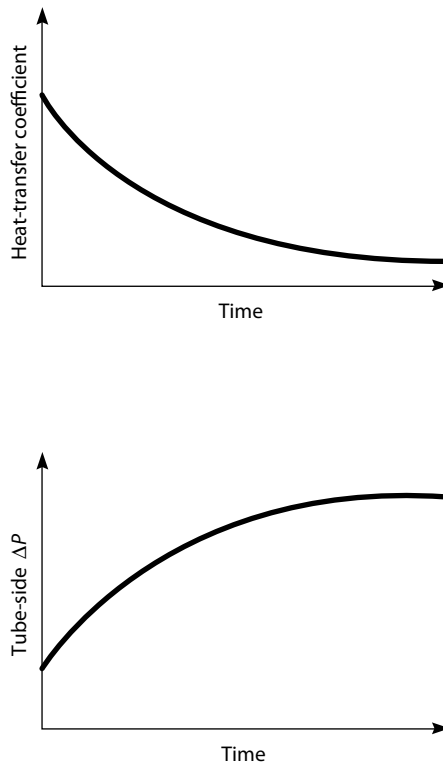
The hotter, fouled tubes must grow. But their horizontal expansion is constrained by the cleaner, colder tubes. Since the colder tubes do not allow the hotter tubes to grow, the hot tubes bend. This, then, is the origin of the twisted tubes we see when an improperly designed tube bundle is pulled from its shell during a turnaround.

The tendency to foul tubes might be eliminated by removal of salts, wax, particulates, corrosion products, polymers, free radicals, and all the myriad of other factors that contribute to fouling. Possibly—but not likely.

The temperature extremes of a process may occasionally be moderated. For example, we desuperheat steam to an amine regenerator reboiler to slow degradation and fouling with the heat-sensitive amine. We avoid heating di(ethylene glycol) above 360°F to minimize fouling in our glycol dehydration unit water strippers. But in general, our latitude to manipulate process temperatures is very limited.

On the other hand, we can, during the design of an exchanger, select a high velocity to combat fouling. For example, let's consider Fig. 26.5. This is data from an exchanger that was designed to operate

FIGURE 26.5
Effect of terminal-tube velocity on exchanger performance.



with a low tube-side velocity. Note how there is a rapid loss in the heat-transfer coefficient as the tubes foul and plug as a result of low velocity. The loss in heat-transfer coefficient U stops only when the terminal-tube velocity is reached in the unplugged tubes.

Note, also, how there is a rapid increase in the pressure drop in the tube bundle as the tubes foul and plug because of the low velocity. The increase in ΔP stops only when the terminal-tube velocity is reached in the unplugged tubes.

It would seem that it might be best to design exchangers for initially high-velocity and high-pressure drops. It would seem that in practice, if we were to design exchangers for a low ΔP , after a few months fouling would cause a high ΔP anyway. This line of reasoning is valid for both the tube and shell sides of a heat exchanger.

26.2 Heat-Transfer Efficiency

26.2.1 “No Fooling—No Fouling”

This is the title of an important article¹ published many years ago. I had the honor of meeting the author, Mr. Gilmore, several years my senior, in 1965. It could be that his sort of plain-talking, hands-on applied science is a vanishing craft. I hope not. The substance of this famous article was simple. When designing heat exchangers, use the available pressure drop to maintain high velocities through the heat-exchanger equipment. Do not use safety factors in allowing for future pressure drop due to fouling. Use of such safety factors will force the mechanical engineer who designs the exchanger to use lower velocities in the design. Then the feared fouling will occur, and the ΔP safety factor will be consumed.

But suppose we are operating a heat exchanger subject to rapid rates of initial fouling. The start-of-run heat-transfer coefficient U is $120 \text{ Btu}/[(h)(\text{ft}^2(^{\circ}\text{F}))]$. Four months later, the U value has lined out at 38. The calculated clean tube-side velocity is 1.5 ft/s. This is too low, but what can be done?

26.2.2 Multipass Exchangers

It is possible to convert the two-pass tube bundle shown in Fig. 26.1 to the four-pass tube bundle shown in Fig. 26.6. This conversion is effected as follows:

1. The center channel head pass partition baffle is cut out.
2. Two off-center channel head pass partition baffles are welded in place, so that 25 percent of the tubes are above the upper baffle and 25 percent of the tubes are below the lower baffle.
3. Both the channel head cover and the channel head tubesheet (see Fig. 26.1) must be remachined to accommodate the new baffles.

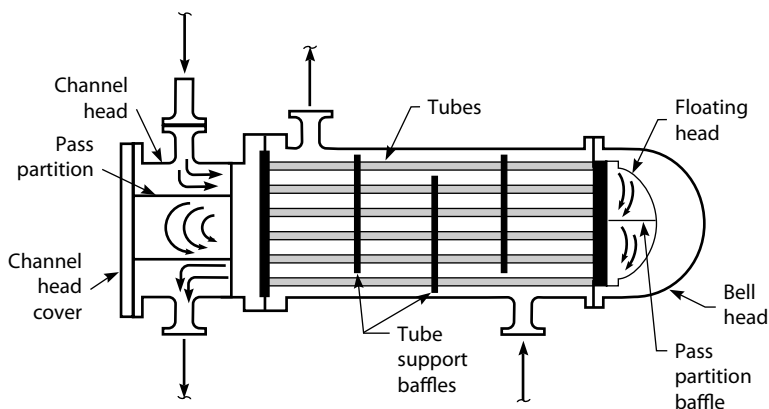


FIGURE 26.6 Four-pass tube bundle.

4. A new, center, pass partition baffle is welded in the floating head. The floating-head tubesheet must also be remachined.

The resulting four-pass tube bundle will have a tube-side velocity twice as high as it did when it was a two-pass exchanger: 3 ft/s. Experience has shown that in many services, doubling this velocity may reduce fouling rates by an order of magnitude. That is fine. But what about pressure drop?

When we convert a tube bundle from two to four passes, the pressure drop increases by a factor of 8. For example, assume that the two-pass ΔP was 5 psig. With the same flow, the four-pass ΔP would be 40 psig. Let me explain:

- Pressure drop increases with the square of velocity.
- If the velocity doubles, the pressure drop would go up by a factor of 4.
- Pressure drop increases directly (linearly) with the length that the flow traverses.
- The fluid must go twice as far in a four-pass as in a two-pass tube bundle.
- Four times 2 equals 8.

Quite likely, even after several years of operation, the pressure drop of a four-pass exchanger will be greater than the ΔP of a two-pass exchanger in the same service. Quite likely, the initial U value of a four-pass exchanger, days after it has been returned to service, will be only slightly higher than the U value of a two-pass exchanger. However, the four-pass exchanger will maintain its U value with

time, far better than will the lower-velocity two-pass tube bundle heat exchanger.

The eightfold increase in pressure drop is certainly a stiff price to pay for this improvement in the long-term U value. But remember this: it is the clean ΔP of the tube bundle that will increase by a factor of 8. Let's say that the tube side of an exchanger is currently operating, after two years of service, in a badly fouled state. Its ΔP , as measured in the field, is 20 psig. The calculated ΔP for clean tubes is 5 psig in a two-pass configuration, or 40 psig in a four-pass configuration. After two years of operation, I would expect the exchanger in the four-pass configuration to have a ΔP of 50 to 60 psig, rather than 160 psig.

Why? Well, because the doubling of the tube-side velocity has promoted turbulence, which retards the accumulation of fouling deposits. Of course, this is exactly the reason why we exercise. The increased flow of blood through our arteries prevents plaque from sticking to the walls of our blood vessels. The plaque, a fatty deposit derived from cholesterol, restricts the flow of blood, which causes high blood pressure and eventually strokes.

26.2.2.1 Erosive Tube-Side Liquid Velocities

Tube-side fouling and plugging is greatly reduced by high velocity. For clean cooling water, a velocity of over 3 ft/s is good. For heavy crude oil contaminated with sediments and salts, a minimum velocity is about 6 ft/s. For cracking unit slurry oil contaminated with catalyst fines, a velocity of less than 5 ft/s will promote tube plugging with catalyst.

The ΔP varies with velocity squared. So there is often a practical upper limit on higher velocities. But, putting aside such hydraulic limitations, there is also a practical limit regarding maximum velocities as limited by tube-side erosion.

When I worked for American Oil, our limit for liquids was 10 ft/s. Recently, working with a very experienced process design contractor, their limit was 14 ft/s. My own design criterion has always been 12 ft/s.

I have worked in plants where I found 12-inch c.s. piping, with piping velocities in corrosive services of 14 ft/s to cause erosive piping failures. For rich amine flow with particulate contamination, piping failures with velocities above 8 ft/s have occurred on units I've worked on.

In summary, for services with moderate corrosion potential, I would keep tube-side velocities below 12 ft/s, and perhaps a bit lower for U tube-type exchangers.

26.2.3 Shell Side vs. Tube Side

We can have any even number of tube-side passes: two, four, six, eight, and so on. But this certainly limits our flexibility to optimize the tube-side velocity.

The shell-side cross-flow velocity may be altered in much smaller increments by changing the tube support baffle spacing. This is one advantage of placing the fluid with the poorer heat-transfer properties on the shell side. But there is another, far more critical advantage in placing the fluid with the poorer heat-transfer properties on the shell side.

26.2.3.1 Laminar Flow

Think about water flowing slowly in a channel. Will the water in the center of the channel flow faster or slower than the water along the sides of the channel? Experience teaches that water in the center of a channel will flow faster than along the sides of the channel. Moreover, if the water flow is really slow, the water creeping along the side of the channel will barely mix with the bulk of the water flowing in the center of the channel. This is called *laminar flow*.

Fluids in laminar flow transfer heat very poorly. The slow-moving fluid creeping along the heat-transfer surface does not particularly mix with the bulk of the fluid. While the slow-moving fluid may get hot, it does not transfer its heat very efficiently to the flow of the bulk of the fluid. This is called *film resistance*.

Laminar flow is very bad for heat transfer. After fouling, it is the second biggest reason for low U values. Laminar flow is caused by two factors:

- Low velocities
- High viscosities

A low velocity for liquids is <2 or <3 ft/s. Velocities of >10 or >12 ft/s may cause erosion of metal surfaces and should be avoided.

A low viscosity is <2 or <3 centipose (cP) or centistokes (cSt). Tap water has a viscosity of about 1 cP. A viscosity of ~ 50 cP is quite high. Warm maple syrup has this sort of viscosity. The viscosity of vapors is almost always very low.

When we cool a liquid off, its viscosity markedly increases. I cannot generalize—it depends on the fluid. But I can say that increasing the viscosity of a fluid from 2 to 40 cP could reduce the observed heat-transfer efficiency (U) from 100 to 25. I know this from my experience in preheating cold, viscous, Venezuelan crude oil, off-loaded from tankers.

The best way to diminish the effect of laminar flow is to place the higher-viscosity fluid on the shell side. The shell side of an exchanger is far more resistant to heat-transfer loss due to film resistance than is the tube side because of

- The vortex shedding shown in Fig. 26.2
- The rapid changes in direction due to the tube support baffles

Purely in terms of heat transfer, it follows that the higher-viscosity fluid should be placed on the shell side. Sometimes, pressure and corrosion force the designer to allocate the higher-viscosity fluid to the tube side. Also, your maintenance department would vastly prefer that the fouling fluid be placed on the tube side.

26.3 Exchanger Cleaning

To disassemble an exchanger may require a lot of work. However, if only the tube side has to be cleaned, the amount of labor required may be halved. Referring to Fig. 26.6:

1. The channel head cover is removed.
2. The bell head (or floating-head cover) is dropped.
3. The floating head is removed.

Each individual tube can now be cleaned by one person with a long tube and a hydroblast machine.

On the other hand, cleaning the shell side requires pulling the entire bundle out of the shell. This is a massive undertaking requiring a special bundle-pulling machine and a large crew of pipe fitters. Then comes the hard part, cleaning the bundle.

If the deposits are soluble in a solvent, the shell may be soaked in a chemical bath. Sometimes this bath must be kept hot to dissolve the deposits at a reasonable rate. Sometimes the deposits must be attacked with a hydroblast machine. If the design engineer has selected a correct *tube pitch*, cleaning the shell side of an exchanger with a hydroblast machine may be a reasonable proposition.

26.3.1 U-Tube Bundles

Figure 26.8 shows a U-tube bundle. Cleaning the U bend of such a bundle is quite difficult even by hydroblasting the tubes with a jet of water. My main objection to U-tube bundles is not the difficulty in cleaning, but the fact that such bundles cannot be inspected after cleaning to verify that the tubes are actually clean. In a floating-head bundle (shown in Fig. 26.6), looking down the straight tubes with the floating head removed allows the verification that the tubes have been completely cleaned.

The disadvantages of the floating-head design (Fig. 26.6) as compared to the U-tube design (Fig. 26.8) are:

- Cost—About 10–15 percent greater
- The difficulty of getting a tight, leakproof closure between the floating-head tubesheet and the head itself

In general, process personnel (like your author) prefer floating heads. The maintenance department prefers U-tube bundles.

26.3.1.1 Effective Cleaning of U-Tube Exchanger Bundles

Hydroblasting U-tube bundles may or may not completely clean the tubes. However, there is a method that is certain to completely clean even a badly plugged and fouled tube—pigging.

Tiny pigs (i.e., small brushes) are individually forced—one pig at a time—through each tube with a high-pressure water gun. One of my clients has cut open U bends to prove the effectiveness of this pigging. It's effective but expensive. About \$10,000 (in 2012) for a crude preheat bundle (3000–4000 ft²). This is a good argument for a floating-head bundle, rather than a U-tube bundle.

There are two types of pigs used. One is a nylon brush and the other is a metal cleaner. At least in cooling water services, the metal cleaner type of pig is the more effective (reference "Steam Surface Condensers," R. E. Putman). This service is provided by Concoindustrial.com.

26.4 Mechanical Design for Good Heat Transfer

26.4.1 Selecting Proper Tube Pitch

From a theoretical heat-exchanger perspective, the triangular tube pitch shown in Fig. 26.7 is best. The term *pitch* refers to the geometry and distance between the holes drilled into the tube-support baffles. A *triangular pitch* means the holes are drilled in the pattern of a 60° equilateral triangle. For example, we might have a 0.75-inch tube (OD) on a 1-inch triangular pattern. The advantages of this tube pitch are

- Many tubes are squeezed into a small shell ID.
- The tube spacing promotes turbulence and hence good heat transfer.

This all results in reduced capital cost. The disadvantage of this tube pitch is that, except for the outer few rows of tubes, the shell side cannot be cleaned by hydroblasting. Obviously, this sort of arrangement should never be used in a fouling service, unless the deposits can be chemically removed.

The second example shown in Fig. 26.7 is 0.75-inch tubes, also set on a 1-inch spacing. But this time, the arrangement is a square pattern. This pattern reduces the number of tubes in a shell of a fixed ID. Also, less turbulence is created by the tubes. Some fluid may flow between the tubes without encountering the tubes at all. All this reduces heat-transfer efficiency. But hydroblasting between the tubes is usually possible—depending on the extent of tube bending and twisting.

The third example shown in Fig. 26.7 is my favorite for sensible-heat transfer in fouling service. This consists of 1-inch tubes set on a 1.5-in rotated square pitch. The pitch layout is the same as that of

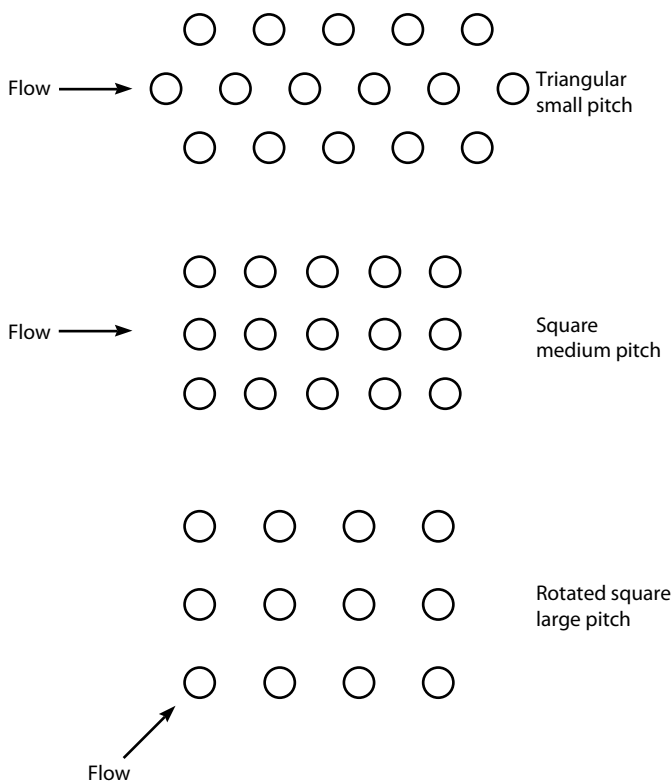


FIGURE 26.7 Square, large-pitch tubes are easier to clean.

the square pitch. It is just the tube bundle that is rotated by 45° relative to the shell-side flow. The shell-side fluid cannot flow without interference between the tubes. Hence, the tubes promote turbulence and improve heat transfer. The large tubes, and especially the doubling of the space between the tubes, reduce the heat-transfer surface area that can fit into a shell of a given ID. But the primary objective of the 1-inch tubes on a 1.5-inch rotated square pitch is that it is relatively easy to hydroblast.

Note that 1-inch tubes on a 1.5-inch tube pitch are 0.5-inch apart. One-inch tubes on a 1.25-inch tube pitch are 0.25-inch apart. The larger spacing greatly reduces the tendency for fouling deposits to bridge the gap between the tubes.

Once the space between adjacent tubes is bridged over with deposits, the flow of fluid between these tubes drops to zero. The local heat-transfer rate drops close to zero. Overall, shell-side ΔP rises as flow is diverted away from the restricted zone.

Using the larger tube pitch will reduce the number of tubes in a shell by about 35 percent. But in a fouling service, this may greatly

increase run times between cleaning and, in my experience, is a worthwhile incremental investment.

26.4.2 Two-Pass Shell

Most of the heat exchangers in your plant are designed with the shell-side inlet and shell-side outlet at opposite ends of the shell, as shown in Fig. 26.6. However, you may have noted a few exchangers in which both the shell-side inlet and shell-side outlet are next to the channel head. This is a two-pass shell-side exchanger of the type shown in Fig. 26.8.

The critical feature of this exchanger is the *longitudinal baffle*, which extends down the length of the shell. The baffle is fitted directly to, and thus becomes physically part of, the tube bundle. The function of the baffle is to force the shell-side fluid to flow down the entire length of the shell and back again to the shell-side outlet nozzle. Such an exchanger is truly countercurrent. The engineer does not apply the F-factor derating coefficient to the log mean temperature driving force, as would normally be used for non-true countercurrent flow.²

This would make it seem that the two-pass shell configuration is inherently more efficient than the ordinary single-pass shell exchanger. And it is, in theory. Then why do we see so few of these exchangers in service?

The reason illustrates the true nature of the shell-and-tube heat exchanger. It is a compromise between an ideal heat-transfer configuration and practical mechanical limitations. In this case, the difficulty is in preventing leakage around the longitudinal baffle. Such leaks permit the shell-side fluid to short-circuit the tube bundle; that is, a percentage of the inlet flow may flow directly to the outlet nozzle. In extremely serious cases, I have seen the bell head (shown in Fig. 26.8) 100°F colder than the shell-side outlet temperature.

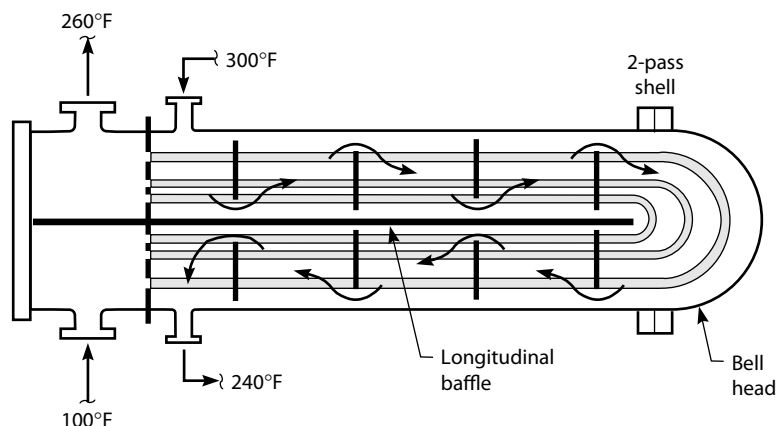


FIGURE 26.8 Two-pass shell exchanger.

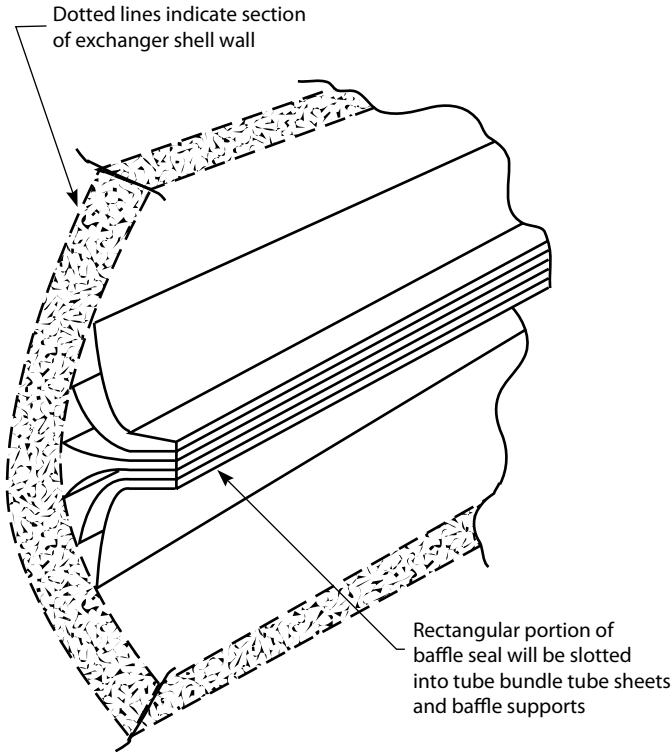


FIGURE 26.9 Detail illustrating how a modern longitudinal baffle seals up against the inside of the exchanger shell.

When new, the longitudinal baffle seal, which is composed of a number of foil strips, extends along the length of the shell. These strips press up against the ID of the shell and effectively seal off the upper half of the shell from the lower half, as illustrated by Fig. 26.9. Unfortunately, the first time the shell is pulled for maintenance, these foil strips are wrecked. Also, unless great care is taken by maintenance personnel when installing the new or replacement longitudinal baffle sealing strips, the heat-transfer efficiency of the exchanger will be greatly reduced after it is returned to service. As a result of this deficiency, most operating companies severely restrict the use of two-pass shell-side heat exchangers.

26.4.3 Double-Pipe Exchangers

Also called “hairpin” exchangers, the double-pipe heat exchanger, as shown in Fig. 26.10, is the simplest of all types we have discussed. It is nothing more than two concentric pipes. The inner pipe might be 2 inch diameter and the outer pipe 3 in. Double-pipe exchangers are

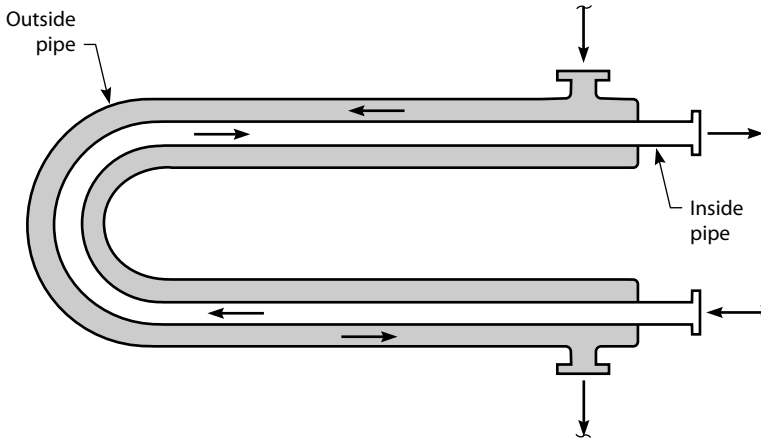


FIGURE 26.10 A double-pipe countercurrent exchanger.

true countercurrent-flow heat exchangers. They are rugged and are reasonably cleanable. Moreover, they are not subject to bypassing. The only problem with them is cost per square foot of heat-transfer surface area. A single pipe or tube simply cannot have much area, typically 100 to 300 ft².

Exchangers of less than a few hundred square feet can be economically designed in a hairpin configuration. Larger exchangers are, for economic factors, shell-and-tube types.

26.4.4 Fin Tubes

You may have seen bundles constructed with serrated, or very small, fins covering the exterior of the tubes. These are called *low-fin-tube bundles*. These fins increase the outside surface area of the tubes by a factor of 2.5. However, this does not mean that the heat-transfer efficiency of the exchanger will increase by 250 percent. Two factors curtail this improvement:

1. Let's assume that the controlling resistance to heat transfer is inside the tube. It does not matter whether this resistance is due to fouling, high viscosity, or low flow. The heat flow must still pass through the smaller surface area inside the tube. The smaller area acts as a choke point that restricts heat flow. Let's assume that 90 percent of the resistance to heat transfer is inside and 10 percent outside the tubes. The overall increase in the heat-transfer capacity of the exchanger would, at best, be roughly 15 percent.
2. Actually, retrofitting a tube bundle with low-fin tubes often reduces heat-transfer capacity. This happens when the

controlling resistance to heat transfer is shell-side fouling. The fouling deposits get trapped between the tiny fins. This acts as an insulator between the shell-side fluid and the surface of the tubes. In severe shell-side fouling services, I have replaced fin tubes with bare tubes, and doubled the heat-transfer duty on the exchanger.

In summary, low-fin tubes may be used to advantage only when the controlling resistance to heat transfer is the shell-side fluid itself.

26.4.5 Heat-Transfer Resistance

The resistances of electric resistors arranged in series may be added to obtain the overall circuit resistance. This concept also applies for a heat exchanger. The following five resistances to heat transfer are added together to obtain the total resistance to heat transfer:

- The tube-side fluid
- The fouling deposit inside the tubes
- The metal wall of the tube
- The fouling deposit around the outside of the tube
- The shell-side fluid

The sum of these five factors is called the *overall resistance* to heat transfer. The reciprocal of the overall resistance is termed U , the overall heat-transfer coefficient.

26.5 Importance of Shell-Side Cross-Flow

Shell-side flow should be mostly at right angles or perpendicular to the tubes. Unavoidably, as the fluid flows from the inlet nozzle to the outlet nozzle, there is some component of flow parallel to the tubes. The bigger the tube support baffle spacing, the greater the component of the flow parallel to the tubes. A larger baffle cut also increases the component of the flow parallel to the tubes. Thus, to promote perpendicular flow to the tubes or cross-flow, velocity baffle spacing should be about 20 to 30 percent of the bundle diameter. The baffle cut should be about 25 to 35 percent of the bundle diameter. By baffle cut, I mean the cutout section of the round baffle.

Reducing the baffle spacing and the baffle cut increases the shell-side pressure drop. But that's the price we must pay for improved heat-transfer rates.

I've read about the importance of shell-side cross-flow in heat-exchanger design books. Higher cross-flow velocities reduce film resistance and promote increased heat-transfer rates. Recently, I had a dramatic personal experience to support this engineering principle.

A refinery in Durbin, South Africa, had a debutanizer reboiler with an observed heat-transfer coefficient of 25 Btu/hr/ft²/°F. The reboiler was a horizontal, thermosyphon, shell-and-tube exchanger. The heating medium was 400-psig steam on the tube side. Except for a short length of the tubes, the shell-side flow was parallel to the tubes. I designed a new bundle (using the existing shell and nozzles) with new tube support baffles. I mostly eliminated the parallel component of shell-side flow. The observed heat-transfer coefficient improved to 80 Btu/hr/ft²/°F. The increased thermosyphon driving force resulting from the larger reboiler duty and vapor generation rate offset the increased pressure drop caused by the new baffles. This kept the circulation rate through the shell constant.

Any time I find that basic engineering principles actually apply in practice, I am always pleasantly surprised. I hope for the best, but plan for the worst. My fear of failure is a basic ingredient in good engineering design.

26.5.1 Detecting Tube Leaks On-Stream

Liz and I were recently working a job at a refinery in Illinois. The pumparound trim cooler steam generator for the fractionator was thought to be leaking water into the pumparound return hot gas-oil flow.

To verify the leak, I proceeded as follows:

- Step 1—On the gas-oil side of the pumparound, I partially closed the outlet valve. This forced the upstream control valve to open.
- Step 2—On the steam side, I reduced the steam generator pressure by partially bypassing the downstream superheater coil.

These steps reduced the steam generator pressure from 170 to 150 psi and increased the pumparound gas-oil inlet pressure from 140 to 160 psi. Liz then checked the blowdown drain on the steam generator and found it to be contaminated with the greenish gas oil. This proved that the steam generator was indeed leaking. Incidentally, there were no tube leaks. The leaks were where the tubes were rolled into the tubesheet. These are called *roll leaks*.

There are a wide range of other techniques used to determine, without taking a heat exchanger offline, that the exchanger tubes are leaking:

- For a hydrotreater feed-effluent exchanger, inject dye into the feed. This dye is used in refineries to identify products for sale. Add the dye at its normal concentration as used for product sales. The dye will be destroyed in the hydrotreater reactor. But, if there is a tube leak in the feed-effluent exchanger, the hydrotreated feed will still be colored.

- For a steam turbine and vacuum-surface condenser, check for dissolved oxygen in the steam condensate. The oxygen would come from cooling water leaking into the shell side.
- For amine regenerator reboilers, at Amoco Oil we used a commercially available leak detector kit. Sulfur hexafluoride was injected into the reboiler steam and the reboiler vapor outlet was then monitored for SF_6 . The detector is sensitive to 1 ppb (billion). The commercially available unit I'm familiar with is sold under the trademark of Fluortracer Analyzer. Both the analyzer and a SF_6 dispenser are needed.

I've also developed my own technique to detect heat exchanger leaks. I'll use the ordinary air conditioner Freon, which is available in most refineries, and an \$800 Freon leak detector kit I've purchased from my neighborhood air conditioner supply store. It's really the same as the SF_6 technique. One of my clients has used a lithium bromide leak detection kit.

References

1. G. H. Gilmore, "No Fooling—No Fouling," *Chemical Engineering Progress*, July 1965, vol. 61, no. 7, pp. 50–56.
2. See TEMA Data Book for F-factor chart.

CHAPTER 27

Heat Exchanger Innovations

We have discussed basic concepts of heat exchanger and air cooler design and operation in earlier chapters. From time to time various innovative ideas to improve the basic shell-and-tube exchanger or air cooler design have been developed. During our seminars, we are often asked for comments on the feasibility of using one or other of these different designs. Therefore, dear reader, in this chapter we offer you our comments regarding a variety of attempts at heat exchanger and air cooler design innovation that have been developed over the last 50 years or so, such as:

- Smooth high alloy tubes
- Low-finned tubes
- Sintered metal tubes
- Spiral heat exchanger
- Tube inserts
- Twisted tubes
- Helical tube support baffles

Caution: Whereas all the designs we include have found some successful applications, the key to success here is to find the right application, and as ever we will add the further cautionary note that the best approach is most often to *copy* a successful application or design rather than to innovate. Therefore, dear reader, please use caution when attempting to use any innovative designs or applications where they have not already been used with success. Ask around if you have an idea to improve heat exchange on your process unit; see if you can find someone else who already had success with the same approach.

27.1 Smooth High Alloy Tubes

Use of smooth high alloy tubes in otherwise corrosive service has been common practice since the 1950s. Tube metallurgy is selected to minimize corrosion while in service. The tubes therefore maintain their smooth, shiny finish. Without sites for fouling deposits to attach or accumulate, the clean heat-transfer coefficient is maintained. This idea has been used with success in refinery catalytic cracker feed preheat service, placing slurry oil on the tube side and vacuum gas oil on the shell side.

Caution: To prevent galvanic corrosion of the tube bundle, the *entire* tube bundle should be constructed of the same metallurgy; this would include the tube support baffles, tube sheets, and tie rods.

We also note that while there is an obvious advantage to maintaining smooth, shiny tubes for applications in sensible heat transfer, this may not be true if it is necessary to vaporize liquid in the exchanger. Some of our clients have experienced difficulties due to film heat-transfer resistance and lack of nucleate boiling sites when attempting to use smooth tubes to vaporize relatively pure component liquids. This would be true for high alloy tubes or simply new replacement bundles for bundles of lesser metallurgy.

27.2 Low-Finned Tubes

The low-finned tubes are made by machining grooves into the exterior surface of the tubes, which increases the surface area by 2.5 times.

The use of low-finned tubes is not a new development. It is an idea that was in common use in the 1960s with mixed results. A successful application is the use of low-finned tubes in clean propane refrigerant condensing service, with *clean* cooling water on the tube side. Observed heat-transfer rates increased by about 70 percent.

However, if the controlling resistance to heat transfer is on the tube side, the increased outside surface area of the tubes does not improve heat-transfer rates.

Caution: If the shell side is subject to fouling, then dirt gets between the fins, in which case the heat transfer will now be worse than with smooth (unfinned) tubes.

27.3 Sintered Metal Tubes

The development of sintered metal tubes dates from the 1970s. A rough coating is applied to the exterior of the smooth new tubes. The hottest fluid is on the tube side. Boiling is intended to take place on the shell side or exterior of the tubes. This is only useful in flux-limited situations when boiling relatively pure components and when heat-transfer rates exceed 10,000 Btu/hr/ft². In such circumstances, the sintered metal tubes act like boiling stones by the way in which they provide sites to initiate nucleate boiling and thereby

overcome flux limitations. The application of sintered metal tubes is limited and in some applications they are either unnecessary, not beneficial, or really should not be used. For example, *do not* use sintered metal tubes in the following applications:

- If exterior of tubes are slightly corroded and pitted, then there are already plenty of nucleate boiling sites
- If a large portion of duty is in the form of sensible heat rather than latent heat
- If fouling on the shell side is significant

27.4 Spiral Heat Exchanger

The spiral heat exchanger concept represents a completely different type of construction as compared to heat exchangers or coolers with tubes or tube bundles. In this very different design, there is only one channel per medium, thus the design aims to eliminate the possibility of channeling and suppress fouling. The objective of the design was to produce an exchanger suitable for heating or cooling of slurries and other process streams containing solids and/or fibers.

Spiral heat exchangers are constructed by winding two long strips of plate metal around a center to form a spiral body, which then contains two concentric channels. Each channel is welded closed at alternate sides (see Fig. 27.1). Covers are then bolted over each side of the spiral body to complete the unit. When both of these covers are removed, the entire surface area of the exchanger is available for manual cleaning.

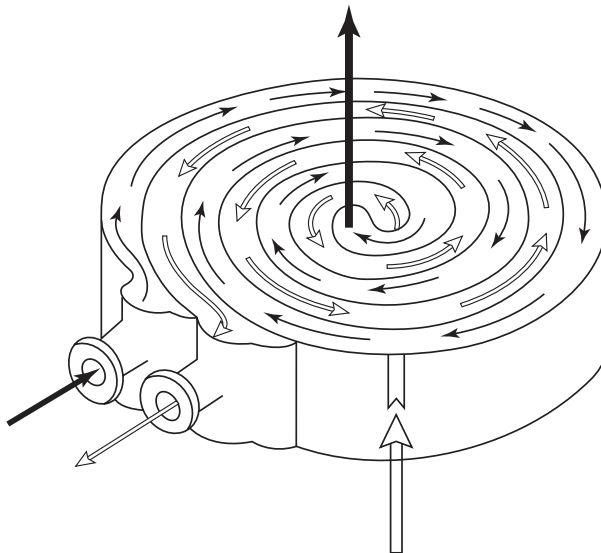


FIGURE 27.1 Spiral heat exchanger cross-sectional flow diagram.

Spiral heat exchangers have been used with success in refinery fluid catalytic cracking (FCC) unit service to cool the main fractionator bottoms product, where they have proved to be less subject to fouling than other types of exchangers. Although one client commented that they had needed to make awkward repair welds between internal channels.

In general, we are informed that this type of exchanger can be constructed of any material that can be cold formed and welded, such as carbon steel, stainless steel, high nickel alloys, or titanium. We are also informed that this type of exchanger may be used at design pressures as high as 430 psig or in vacuum service, and at temperatures as high as 750°F or as low as -40°F.

This type of exchanger is produced by AlfaLaval.

27.5 Tube Inserts

We are aware of three different kinds of tube inserts, but all three are based on the idea of a metal spiral inside the tube. The commercial names are:

- Spirelf
- Turbotal
- Fixotal

Spirelf was originally developed by Elf Research whereas Turbotal and Fixotal were originally developed by Total Research. Spirelf and Turbotal are the products of independent research programs launched by Elf and Total in the 1980s with the primary aim being to reduce fouling in crude preheat exchanger trains. All three of these tube inserts are now marketed by Petroval (see Fig. 27.2).

27.5.1 Spirelf Tube Insert

The Spirelf tube insert is a metallic coil stretched over the length of the tube and held in place at each end by straight wires. In operation, when fluid flows through the tube, the Spirelf coil vibrates axially and radially.

The concept is that this vibration reduces fouling and increases flow turbulence, producing an improvement in the heat-transfer coefficient. However, there is also an associated increase in tube-side pressure drop of about 3 psi per pass at tube-side velocities of 3 ft/s.

The main applications for Spirelf so far have been oil refinery preheat trains for CDU, VDU, FCC, water treatment feed exchanger HDS units, and also high-tube velocity applications as found in reboilers.

27.5.2 Turbotal Tube Insert

The Turbotal tube insert coil, also referred to as a helicoid mobile, is designed to rotate within the tube. This is achieved by a mechanism at the tube inlet that holds the Turbotal insert in place but allows it to

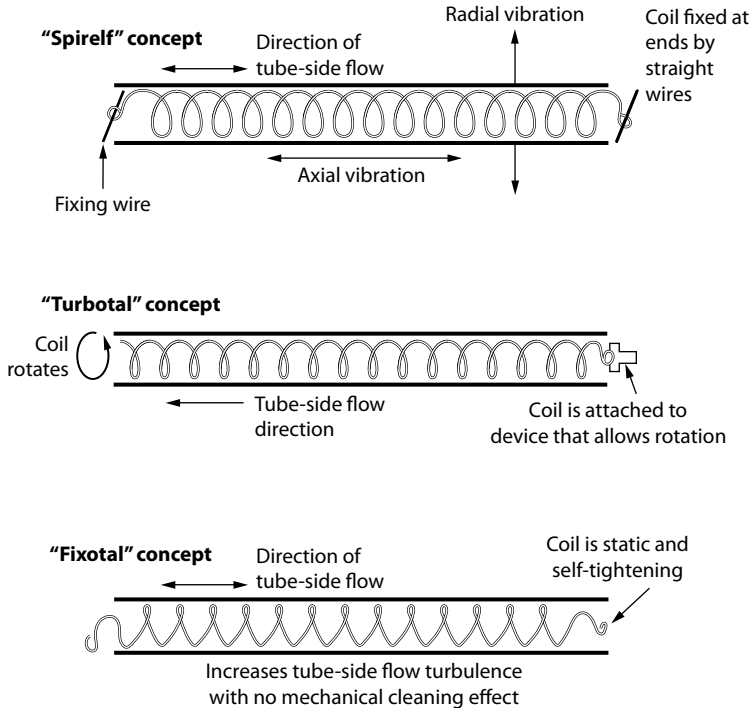


FIGURE 27.2 Tube inserts: Spirelf, Turbotal, and Fixotal.

rotate. The velocity of the fluid in the tube determines the speed of rotation. There is also some micro-vibration of the coil within the tube.

The concept is that the tube-side fouling rate will be reduced by continuous scrubbing of the tube wall and also have a favorable impact on heat transfer. The associated increase in tube-side pressure drop with this device is about 1.5 psi per pass at 3 ft/s tube-side velocity, about half that of Spirelf or Fixotal.

The main applications for Turbotal so far have been in the pre-heat trains of oil refinery CDU, VDU, FCC units, and water treatment exchangers. Maximum tube-side velocity up to 7 ft/s, but can not be back flushed or sustain reverse flow for cleaning as that would un-hook the turbotal coil.

27.5.3 Fixotal Tube Insert

The Fixotal tube insert is a metallic coil, which is static and self-tightening inside the tube. There is no intended mechanical cleaning effect with this insert.

The concept is to increase turbulence in the tube-side flow, especially at the tube wall by constantly mixing slow moving fluid near the tube wall back into the bulk, thus increasing tube-side heat-transfer rate and reducing fouling. The additional tube-side pressure drop

with the Fixotal insert is about 3 psi per pass at tube-side flow velocities of 3 ft/s.

The Fixotal inserts have been used in single- and two-phase flow applications—that is, condensation and evaporation. Only the Fixotal inserts are used in air coolers and are intended to reduce fouling, particularly with paraffin coating or water cooler scaling. We also note that Fixotal has been used in U tubes, and can now be left in place during hydroblasting for cleaning.

27.5.4 General Comments on Spirelf, Turbotal, and Fixotal Tube Inserts

In some of the early Spirelf and Turbotal installations, although there was initial reduction in fouling rates, there were also problems with mechanical integrity of the tube inserts; broken pieces of the tube inserts were all too often found in pass partition boxes of the heat exchangers. We have been informed that improvements to the design of Spirelf and Turbotal were made to combat these problems and apparently the end results are favorable, although we note that the life expectancy of the Turbotal inserts may be a two year maximum, whereas Spirelf and Fixotal have up to five years' service time.

Naturally these tube inserts may be attractive as retrofit items, with potential savings in energy costs and reduction of downtime for exchanger cleaning, but it is important to be careful when selecting these, as with any piece of equipment, to be sure that the tube inserts are appropriate for the service for which you intend to use them. Note that both Spirelf and Fixotal can withstand back flushing for cleaning but Turbotal can not.

Petroval now has over 20 years' experience with the design, installation, and operation of these types of tube inserts. There is also standard service offered by Petroval from initial valuation to select the most favorable type of insert to suit clients' working conditions, economic evaluation of projected savings, insert manufacture, super-vision or (if preferred) management of installation, and monitoring of performance.

When these inserts work well, they seem to work very well, but sometimes they are selected for an unsuitable application, just as in the case of the particular air cooler that we describe below. It is clearly most important to carefully evaluate the suitability of the service that you intend to retrofit, consult and work closely with manufacturers of the equipment, ask around and seek out other similar applications to compare performance of the retrofit in the field. You cannot fight heavy fouling service with these inserts. The metallurgy of coil and corrosivity of fluid must also be considered.

27.5.5 Air Cooler Retrofit

In a used motor oil reprocessing facility, the vacuum tower precon-denser consisted of a large fin-fan air cooler. Inlet conditions were 260°F and 80 mm Hg. Due to the contaminants in the used motor oil,

this is a potentially very fouling service. Tube inserts, called turbulators (basically wire springs), were installed in the finned tubes by the manufacturer. The initial performance was reported by our client to be good. However, 18 months later we observed a high-pressure drop (20 to 30 mm Hg) through the air cooler vacuum precondenser. Our client reported to us that after our visit, they had attempted to:

- Clean the tubes by hydroblasting, but could not establish a water flow through the tubes.
- Extract the wire spring turbulators from the tubes, but without any success. That is, the springs were stuck.

The air cooler bundle was then replaced with an ordinary finned tube bundle. The use of wire inserts in fouling service, where extra pressure drop is totally unacceptable, represents poor engineering design practice. This is especially so when the controlling resistance to heat transfer is going to be on the air side of the precondenser cooler.

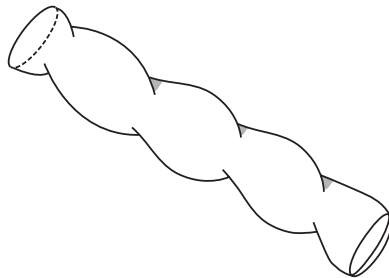
27.5.6 Concluding Thoughts on Wire Spring Turbulators

Wire spring turbulators may have an advantage over tubes with no inserts in that they keep the tubes cleaner. This seems to work best to prevent fouling if it is as a film. If fouling is in the form of particulates in the feed, then the tubes have a tendency to plug. There is also the effect of increased pressure drop to be taken into account when considering tube inserts as a retrofit. However, the decrease in temperature gradient between the tube wall and the bulk fluid may work well to reduce fouling when the fouling is tube-wall temperature dependent. This was seen particularly on a furfural unit when a fixed tube insert was used on the rundown cooler and reduced the storage temperature by about 54°F.

27.6 Twisted Tubes and Twisted Tube Bundle

The concept starts with the use of a tube that is itself twisted, the idea being to promote turbulence within the tube as a result of the tube wall shape. When you look at one of these tubes, the shape is like a large hollow drill bit (see Fig. 27.3). These twisted tubes are then

FIGURE 27.3
Twisted tube looks like a hollow drill bit.



arranged in such close triangular pitch on the tube sheets that they are actually touching each other at many points along the length of the tubes (see Fig. 27.4). One might think that to have the tubes touching would cause a problem with shell-side fouling, but the manufacturers claim that because the points of contact occur on the curved ridges of the tubes, rather than being a problem the points of contact actually increase shell-side turbulence.

The manufacturers also claim that because of all these points of contact they are able to avoid the use of baffles for tube support and therefore eliminate problems of vibration. (Tube vibration in traditionally designed shell-and-tube exchangers particularly at the point where the tubes pass through the tube support baffles can cause the tubes to fail and leak.)

The other basic concept that is essential to the functioning of a tube bundle made with these twisted tubes is that in manufacture of the bundle, the tubes have to be grouped or lined up in the bundle so that the ridges do meet and touch to form the curved points of contact described above. Also and most significantly, this alignment

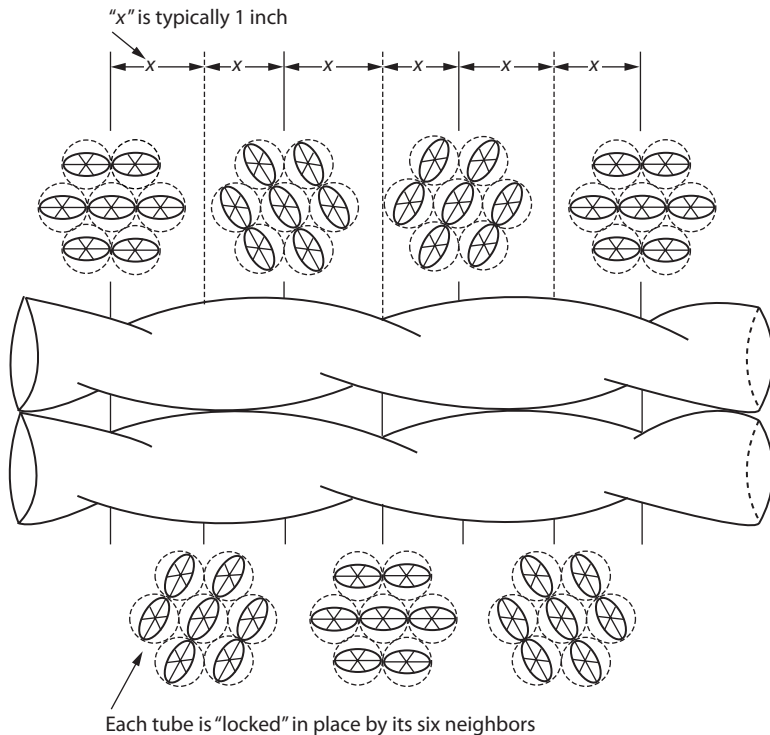


FIGURE 27.4 Points of contact for twisted tubes on close-packed triangular pitch.

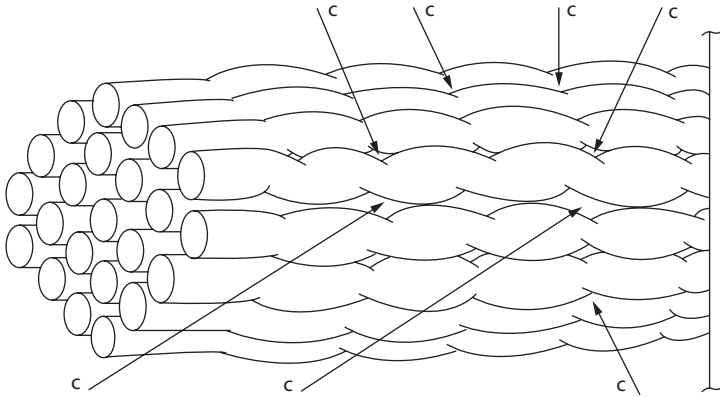


FIGURE 27.5 Simplified portion of a twisted tube bundle indicating cleaning lanes “c,” tubesheet not shown.

produces shell-side “cleaning lanes” at various angles all the way through the bundle between the ridges on the tube surface (see Fig. 27.5). Various advantages are claimed for these twisted tube bundle exchangers:

- Reduced fouling due to increased turbulence on both shell side and tube side.
- Increased heat transfer as a result of increased turbulence and lower fouling rate.
- Reduced pressure drop as compared to traditional shell-and-tube heat exchangers both shell side and tube side.
- Increased heat-transfer capacity as a result of around 30 or even 40 percent increase in surface area as a retrofit bundle in an existing shell. This extra surface area is achieved because the tubes are so closely packed.
- Elimination of the risk of vibration as a result of each tube being tightly locked into place by its six immediate neighbors, and also allowing greater velocity than traditional shell-and-tube exchangers.
- Use of a shroud around the bundle inside the shell allows the use of existing inlet and outlet nozzles. (A shroud that incorporates an impingement plate can be made in two pieces that bolt together for ease of removal during bundle cleaning operations.)
- Smaller shell size on new equipment as compared to traditional shell-and-tube exchangers.

We have heard that these exchangers have been used in a variety of oil refinery applications both in horizontal and vertical exchangers, some of which include:

- Semiregenerative naphtha reformer feed effluent exchangers, where reduced pressure drop brings additional benefits, increasing the hydrogen recycle ratio without changes to the recycle gas compressor
- Hydrocracker
- Hydrotreating (feed-reactor effluent exchangers)
- Alkylation units, both on sulfuric acid (feed-reactor effluent exchangers) and hydrofluoric acid units
- Crude preheat train, with crude on tube side or sometimes on shell side (see next section)

27.6.1 Twisted Tubes Are Used in Crude Preheat Mainly with Crude on the Tube Side

The main benefit that is found seems to be the increase in surface area, typically 40 percent more surface area in the bundle due to the fact that there are more tubes in the bundle compared with a conventional bundle that would fit the same shell. Actual experience shows that the twisted tubes may foul more readily and will plug if fouling deposits are from particulates in feed. But the twisted tube design does work in this service to combat fouling if deposits are formed as a fouling film inside tubes.

Special techniques are required for tube-side cleaning, but cleaning the tube side is not any more difficult than cleaning conventional tubes.

At one Midwest refinery, on a 180,000 BSD crude unit, two parallel, 6000 square ft, titanium twisted tube bundles have been in service for a number of years. Crude tower overhead vapors plus steam are condensing on the shell side. Crude is on the tube side. Fouling on both the shell side and the tube side appears to be quite minimal. However, the heat-transfer coefficient is bad; only about 20–25 Btu/hr/ft²/°F, even when the exchanger is clean.

We attribute this mainly to the relatively low (3 ft per second) tube-side velocity and cold (50°F), somewhat heavy (25°API) crude with a high viscosity. The overall performance of these exchangers did not improve all that much, according to our client, after the new twisted tube bundles were installed. However, they also converted the exchangers from four to two tube-side passes, and this decreased the tube-side velocity by half. Presumably, the reduced tube-side velocity offset the benefits of the twisted tube design.

Converting back to four passes is very difficult, because the tube ODs are so tightly packed with twisted tubes, and there is very little space to accommodate additional new pass partition baffles.

27.6.2 Twisted Tubes with the Crude on the Shell Side

We have heard of a few applications of this. We note that if fouling deposits are particulates in the feed, this appears to be a bad application.

27.6.3 General Problems and Concerns with Twisted Tube Exchangers

The main concerns seem to be, "Is it going to plug up, and are we going to be able to clean it?" The answer is that in general people find it is possible to clean both the shell side and tube side following manufacturer's guidelines (time period and method) and using fairly standard cleaning equipment. However, there have been instances when cleaning has been difficult or even impossible.

Special care should be taken on acceptance of a new twisted tube bundle to verify that the tubes are lined up correctly and that *all* shell-side cleaning lanes are unobstructed.

We know that although there is concern as to how to clean these twisted tube bundles, in some cases, the run length has been so long that even years after installation there still has been no need to clean the bundle. Apparently when it works well, it works very well.

Other problems and concerns with twisted tube exchangers have been:

- The need to take extra care when pulling and reinserting the bundle because, especially with the surrounding shroud, twisted tube bundles are less robust than traditional shell-and-tube exchanger bundles.
- The need to take more care when commissioning twisted tube bundles in very hot service because they are less rugged than traditional shell-and-tube exchanger bundles, and there may be a greater risk to tubes being pulled out from the tube-sheets by uneven thermal expansion.
- The cost of the twisted tube bundle as compared to a new larger traditional shell-and-tube exchanger of comparable surface area with wide rotated square pitch for ease of cleaning. In some cases the traditional shell-and-tube exchanger (shell + bundle + installation) was the least expensive option.
- Putting crude on the shell side of a twisted tube bundle may cause problems with fouling. Do not put vacuum resid on the shell side with twisted tubes.

Twisted tube bundles are manufactured by a division of Koch-Glitsch and are also marketed by Brown Fintube in Luxembourg and the United Kingdom.

27.7 Helical Tube Support Baffles

Tube bundles with helical baffles are now produced by both ABB Lummus Heat Transfer and Koch-Glitsch. HTRI now has a rating for this type of design option in their software.

Although the traditional shell-and-tube heat exchangers have segmental tube support baffles at 90° to the line of the tubes, the helically baffled tube bundle has tube support baffles sloped so as to induce a helical flow pattern on the shell side (see Fig. 27.6).

The concept is to ensure effective bundle penetration and avoid dead zones so as to leave no stagnant areas where shell-side fouling could accumulate. Also, helical baffle plates can be spaced to reduce unsupported tube spans (without affecting pressure drop or heat-transfer characteristics) to mitigate flow-induced vibration. The designers also claim that the helicoidal baffle arrangement permits increased heat transfer.

We have heard of helical baffle exchangers being used in oil refineries to preheat delayed coker feed with vacuum resid on the shell side (with the helical baffles). In this service, the helical baffles seem to work well, resulting in less fouling, slower rate of increase in

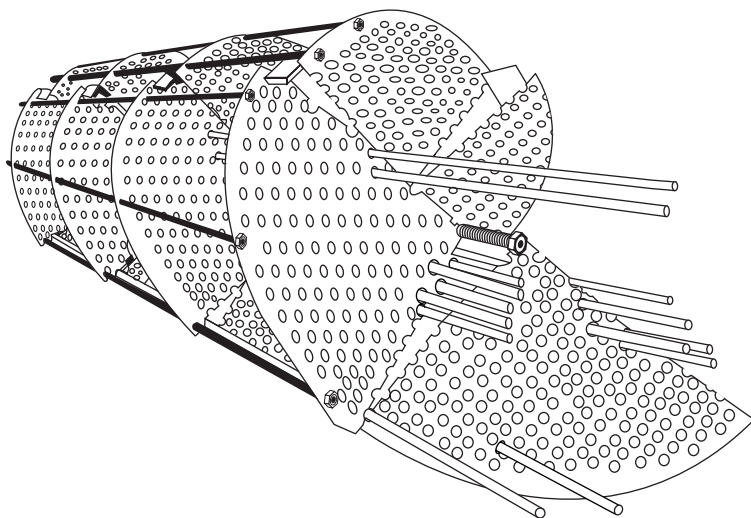


FIGURE 27.6 Schematic arrangement showing helical tube support baffles in relation to tube alignment.

pressure drop, and better maintenance of heat-transfer coefficient as compared to traditional shell-and-tube exchangers. These effects have proved consistent over several years.

Another example from oil refineries is in crude preheat service with vacuum resid again on the shell side (with the helical baffles). Once again, similar results are seen as described above, with less fouling, reduced rate of increase in pressure drop, and better maintenance of heat-transfer coefficient as compared to the conventional shell-and-tube exchanger design. Our more direct experience of this comes from current practice in the United States, but we have also seen evidence of similar applications in Australia, as discussed in a recent article on the subject of crude preheat exchanger train redesign.¹

We have also heard of a case where a heat exchanger was retrofitted with a helical baffled bundle and the shell-side fouling actually increased. The client theorized that being as how the angle used to slope the baffles can apparently be varied by the designer according to service conditions, the angle for the baffle had not been correctly selected.

In general, helical baffle tube bundles are most cost-effectively used as replacement bundles in existing shells. However, for this to actually improve heat-transfer rates, the controlling resistance to heat transfer must be on the shell side and not the tube side of the heat exchanger. Basically this means that the shell-side flow must suffer from some combination of:

- High viscosity
- Low velocity
- High rates of fouling
- Low available pressure drop

For an equivalent shell-side ΔP , the helical baffle will be less fouling and will provide less film resistance, and less fouling resistance than will a conventional vertical tube support baffle.

In summary, the helical baffle design will not help improve the performance of a shell-and-tube heat exchanger, if the controlling resistance (fouling or film) is on the tube side.

Reference

1. D. Singh and S. Van Wagensveld, "Redesign Crude Preheater Train for Efficiency- Over Design of Heat Exchangers Sourced to Fouling Problems," *Hydrocarbon Processing*, May 2007.

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CHAPTER 28

Shell-and-Tube Heat Exchangers: Design Details

I will only be discussing details for an ordinary shell-and-tube exchanger, assuming liquids on both the shell and tube sides. Condensers and reboilers will be discussed separately.

There are three sorts of exchangers that are in common use:

1. Pull-through (or split-ring) floating head
 2. U-tube bundle
 3. Fixed tubesheet (i.e., both ends of the tube bundle are permanently fixed to each end of the shell)
-
1. Fixed Tubesheet—The advantage of a fixed tubesheet exchanger is that it is a true countercurrent flow exchanger and no reduction of the log mean temperature difference (LMTD) is required for non-true counter-current flow between the shell-side fluid and the tube-side fluid. However, it's impossible to mechanically clean the shell side, as the bundle cannot be extracted from the shell. When such an exchanger fouls on the tube side, it can easily be cleaned. When it fouls on the shell side, you can throw the entire exchanger away.
 2. U-Tube Bundle—The tube side of a U-tube bundle can be cleaned, including the return bends. The problem is not tube-side cleaning, but tube-side inspecting. There is no practical way to see if the tubes have actually been hydroblasted clean. While a U-tube exchanger is less expensive than a conventional floating-head exchanger (10–15 percent), it should not be used if tube-side fouling is a factor.
 3. Floating-Head Bundle—Designated as an “AET” pull-through floating-head exchanger (or “AES” floating with a split-ring backing device), this sort of a bundle has one fixed tubesheet

and one floating tubesheet. One end is left free to float to accommodate the differential rate of thermal expansion between the shell side and the tube side. In this exchanger, the tubes are straight and can immediately be inspected to see if they have been properly cleaned. However, unlike the fixed tubesheet configuration, this floating-head bundle is not a true countercurrent flow configuration. Hence, its LMTD must be reduced by the F -factor correction factor for non-true countercurrent flow.

The floating-head exchanger is quite a bit more expensive to purchase than the U-tube or fixed tubesheet exchangers. Also, your maintenance department will generally dislike making the gasketed closure between the floating-head cover and the floating-head tubesheet. Finally, it's not uncommon for this closure to leak when the tube-side pressure exceeds the shell-side pressure. Still, the "AES" or "AET" exchanger is typically the correct design for almost all applications.

28.1 Selecting the Process Fluid Location

28.1.1 Shell or Tube Side

The fluid with the high viscosity belongs on the shell side. The objective is to avoid laminar flow and high heat-transfer film resistance. Typically, the higher viscosity fluid has the greater fouling tendency. Thus, the normal practice is to place the dirty, higher viscosity fluid on the tube side for ease of cleaning. However, if the shell side is designed correctly, it is quite straightforward to clean. I'll detail the design criteria below for ease of shell-side cleaning. But first I'll explain the critical parameter called vortex shedding.

28.1.2 Vortex Shedding

When fluid flows at a 90° angle across a tube, it creates turbulence, which is called *vortex shedding*. The swirls created when a river flows across a tree stump is a common example. The swirls destroy the film resistance to heat transfer created by laminar layers of fluid surrounding the tube. My experience indicates that a shell-side cross-flow velocity of 3 to 5 ft/s is sufficient to obtain this benefit.

To calculate the cross-flow area, determine the open area between the tubes at the edge of the tube support baffles. For example:

- Baffle spacing = 10 inches
- Number of tubes at edge of the tube support baffle = 20
- Tubes are 1 inch on a $1\frac{1}{2}$ -inch center, or $\frac{1}{2}$ -inch space between each tube

Therefore, cross-flow area =

- $(10 \text{ inches}) \times (20 \text{ tubes}) \cdot (\frac{1}{2} \text{ inch}) = 100 \text{ square inches} \div 144 = 0.70 \text{ ft}^2$ (neglecting gaps at end of tube rows)
- If the fluid flow was $3 \text{ ft}^3/\text{s}$, then the cross-flow velocity would be $3 \div 0.70 = 4.3 \text{ ft/s}$ (this is approximate as I've not accurately accounted for gaps at either side of the row of tubes)

In my book, *Process Design for Reliable Operations*, I've given a vivid example of the terrible consequences of placing high-viscosity industrial fuel oil on the tube side of a water cooled exchanger.

The second reason that shell-side flow is not subject to laminar flow is that the liquid only flows the diameter (several feet) of the shell, before its direction is reversed by the tube support baffles. That's not far enough to establish laminar flow. In the ordinary range of operations, there is no such thing as laminar flow on the shell side.

28.1.3 Adjusting for Cross-Flow Velocity

The designer ought to specify the baffle spacing to achieve the desired cross-flow velocity. In a 20-ft-long bundle, a minimum of about five baffles are needed to actually support the tubes. The remaining baffles are to induce an adequate cross-flow velocity. I've designed tube bundles for 30 baffles (i.e., 8 inch spacing) with no problems. Just remember that doubling the number of baffles increases ΔP by a factor of 8. Let me explain:

1. Delta P varies with velocity squared (which has doubled)
2. Now multiply by the flow path length
3. $(2)^2 \cdot (2) = 8$

I use a baffle cut of about 25 to 30 percent, but not for any particular reason. It's just that it seems to work out okay.

28.2 Design the Shell Side for Ease of Cleaning

There are two ways to design the shell side of an exchanger for fouling service:

	Stupid	Smart
Tube Size	$\frac{3}{4}$ inch	1.0 inch
Tube Pitch	15/16 inch	1½ inch
Tube Pitch Pattern	Triangular	Rotated square

Using 3/4-inch tubes on a 1-inch rotated square is okay, but not as easy to clean as the smarter method. Of course, being smart costs money. Meaning the smart tube bundle ID will be roughly 20% bigger than the stupid bundle. But that's a small price to pay for putting the higher viscosity fluid on the shell side rather than on the tube side and then suffering from laminar flow.

28.2.1 Tube Materials

One excellent way of suppressing shell-side fouling (and tube-side fouling as well) is to maintain tube smoothness by retarding corrosion. Basically, use alloy tubes (316 s.s. if chlorides and caustic are not present), rather than carbon steel (c.s.).

Don't make the mistake of using c.s. tube support baffles and 316 s.s. tubes. A galvanic cell will be created that will destroy the c.s. tube support baffles. I've seen the result of this error on an HDS unit reactor effluent.

Don't rely on sacrificial anodes. They are not reliable and are typically not replaced in a timely manner.

Construct the entire bundle out of 316 s.s. It's likely two to three times more costly than c.s., but the shell and installation costs (90 percent of the total) will not change. And rates of fouling will be reduced, often quite dramatically.

28.2.2 Tube-Side Velocity Constraints

Exchangers can be any even number of tube-side passes that produce the desired tube-side velocity. Depending on the tendency of the tube-side fluid to foul, a minimum tube-side velocity will be 3 to 5 ft per second.

A maximum is 10 to 12 ft per second to avoid erosion, based on the nature of particulates in the feed.

More importantly, remember to stay well above laminar flow, as defined by Reynolds's number (i.e., above 2000).

$$\text{Re} = \frac{(\text{Tube ID}) (\text{Flow}) (\text{Density})}{\text{Viscosity}}$$

High viscosity being the big factor.

Also, ΔP , as discussed above for the shell-side flow (see preceding section on cross-flow velocity), varies with the number of passes, cubed (not squared). That is, flow path length doubles when going from two to four tube-side passes. Meaning, if I convert a two-pass exchanger to four-pass, the fluid has to flow twice as fast and twice as far.

28.2.3 Terminal Tube Velocity

The effects of excessively low tube-side velocity will be to damage the tube bundle due to terminal tube velocity problems.

When a tube in fouling service (where the colder fluid is on the tube side) begins to foul due to low tube velocities, the flow through the tube is reduced. The tube gets hotter, which further reduces the flow as it fouls quicker. The resulting suppressed velocity then accelerates the rate of tube-side fouling until the tube plugs off entirely.

Now the totally fouled tube heats up to the shell-side temperature, while the tubes that are not plugged remain cooler. The resulting divergence in tube temperature causes the hotter tubes to bend and distort due to the differential rates of thermal expansion between the cold tubes (not plugged) and the hotter tubes (totally plugged).

I see this frequently when tube bundles are extracted from the shell during a unit turnaround. The distorted tubes interfere with the proper fluid flow through the shell side of the exchanger and likely promote both shell-side fouling and shell-side bypassing. Also, as the tubes plug off, tube-side ΔP increases. If half the tubes plug, then the differential pressure across the channel head pass partition baffle will increase by a factor of four and may result in the failure of the channel head pass partition baffle.

The designer should keep these factors in mind when selecting the number of tube-side passes. Selecting too few passes and a low tube-side ΔP is often not a conservative design. Better to err at too high a tube velocity, than too low a tube velocity.

28.2.4 Exchanger Nozzle Sizing

Especially during retrofits, designers often neglect to check that revamped existing exchangers have adequate-size nozzles.

- Tube side—The channel head nozzles need to be roughly as large in area as the cross-sectional area of the tube passes.
- Shell side—The shell inlet and outlet nozzles should be set roughly as large as the cross-flow area as defined above.

This is conservative sizing. But I hate to waste valuable ΔP on parasitic pressure losses across nozzles.

Also, if the piping sizing is larger than my calculated nozzle sizes, I'll match up with the piping. If the piping is smaller than my calculated exchanger nozzle sizes, then I'll retain my larger exchanger nozzle sizes.

28.2.5 Impingement Plates and Seal Strips

The shell-side impingement plate must be a bit larger than the inlet nozzle to protect the upper rows of tubes from the erosive action of the shell-side inlet fluid. To accommodate the impingement plate, several rows of tubes must be omitted along the top of the bundle. The space so created allows the shell-side flow to bypass the entire tube bundle by sneaking across the top of the bundle.

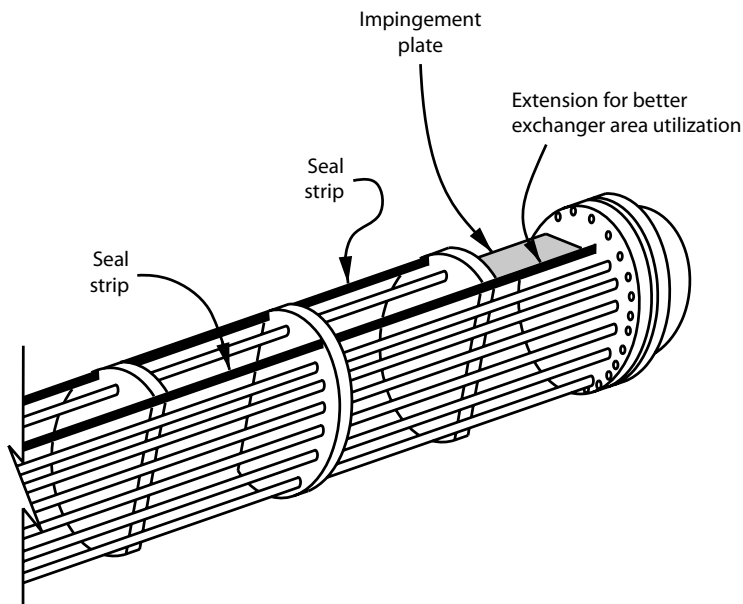


FIGURE 28.1 The seal strips avoid shell-side bypassing.

To dam off this bypass area, a pair of seal strips is needed. These are typically $\frac{1}{4}$ -inch thick, as long as the tubes, and perhaps 3 or 4 inches wide. They are set in grooves cut in the tube support baffles on either side of the impingement plate (Fig. 28.1).

However, when I wrote that the seal strips needed to be as long as the tubes, that is not quite right. One of the pairs of seal strips must have one strip that stops at the first tube support baffle. If not, the shell-side feed would be trapped above the impingement plate.

If we have vertically cut the tube support baffles (which is normal, at least in the United States), then the seal strip that is aligned with the edge of the first shell-side tube support baffle should extend to the tubesheet.

Normal design practice is to end both seal strips at the first tube support baffle. This is wrong design practice. It allows the shell-side flow to bypass the exchanger area between the tubesheet and the first tube support baffle. To avoid this problem, the seal strip at the edge of the first tube support baffle must be extended to the tubesheet (see Fig. 28.1).

28.2.6 Back-Flush Connections—Water Coolers

I prefer to place water cooler back-flush connections on the channel head cover. This is mainly because the piping design is not under my

control, and thus the tube-side back-flush connection may be omitted by the mechanical designer.

My rule of thumb is that I don't want the back-flush connection to be less than half the ID of the cooling water supply line.

You need to give back-flushing a little extra thought if the exchanger is to be located well above grade. Will the cooling water return header pressure be great enough to back-flush the tubes? For example, if the cooling water supply pressure at grade is 30 psig, and the exchanger is 70 ft above grade, then the water return header pressure will be under a vacuum and cannot be used for back-flushing. Obviously, there is no neat answer to this limitation. Use of plant firewater is not advisable, because one should never divert firewater for safety reasons.

28.2.7 Helical Baffles

There are a number of novel heat-exchanger designs that I am familiar with:

- Twisted tube
- Tube inserts
- On-line circulating balls for tube-side cleaning
- Helical baffles
- Spiral heat exchangers

The only one of the above that is in widespread use, and used with success in the correct application, is the helical baffle exchanger (see Figure 27.6). The tube support baffles are set at an angle to induce a helical flow pattern on the shell side. The idea is to avoid the dead zones that occur in a conventional vertical tube support baffle. Thus, ΔP and fouling are both reduced on the shell side.

However, I have three reservations:

1. If the controlling resistance to heat transfer is on the tube side, the helical baffle arrangement serves very little purpose.
2. For a new installation, it's likely cheaper just to buy a somewhat larger, conventional bundle than a proprietary helical baffle bundle.
3. The design angle of the baffles is quite critical to obtain the full benefit of the helical baffle bundle.

The helical baffle bundle is best used during retrofits to existing exchangers, if the controlling resistance to heat transfer is on the shell side.

Spiral heat exchangers are excellent in fouling services. However, after cleaning, they are difficult to reassemble without internal leakage.

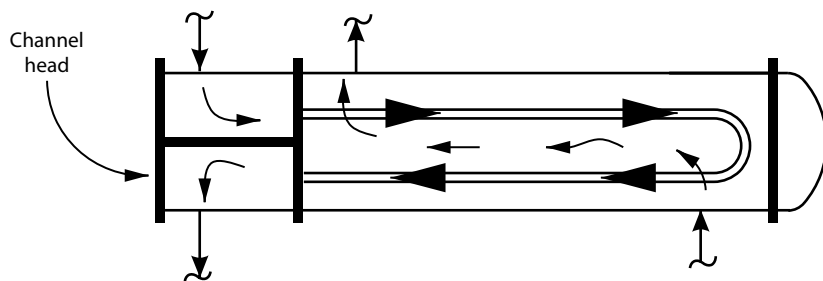


FIGURE 28.2 Top half of exchanger is in countercurrent flow, which is good. Bottom half is in co-current flow, which is bad.

28.2.8 Correction Factor for Non-True Countercurrent Flow

When we calculate the LMTD in a heat exchanger, we assume that true countercurrent flow exists between the shell side and the tube side. For an ordinary exchanger, as shown in Fig. 28.2, this is not true.

As a consequence of the channel head tube-side inlet and outlet being located on the same end of the exchanger, the lower half of the shell is in co-current flow. Depending on the temperature profile, this typically reduces the LMTD by 5 to 25 percent. To calculate this loss in heat-transfer efficiency due to this problem, we use the *F*-factor correction factor as presented in your TEMA Data Book.

If you are unwilling to incur the penalty of loss of heat-transfer LMTD as calculated by the *F*-factor, then you have three choices:

1. Place two exchangers in series on both the shell and tube sides. In general, this is the preferred solution and the one I would adopt.
2. Use double-pipe exchangers. These are indeed true countercurrent flow exchangers (also called hairpins). I often use these. They are reliable and robust, but if the required capacity exceeds several hundred square feet, they are too small for many applications.
3. Use two-pass shell exchangers, as discussed below.

28.2.9 Two-Pass Shell Exchangers

Figure 28.3 shows that such an exchanger is in true countercurrent flow.

The shell-side inlet and outlet nozzles are located on the same side, close to the channel head. The cross-baffle shown (longitudinal) is required to direct the shell-side flow along the length of the shell.

This looks good on paper. But the problem arises in sealing the cross-baffle to the interior of the shell. This is accomplished by means

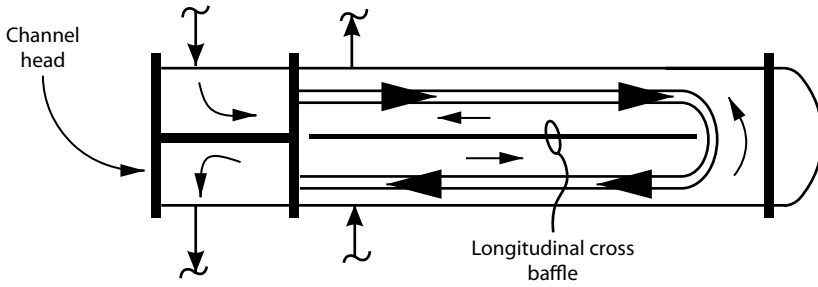


FIGURE 28.3 The longitudinal baffle eliminates co-current flow and increases LMTD.

of seal strips. I'm using the term *sealing strips* in a completely separate way than used in the preceding section of this chapter.

The seal strips, as used in association with the longitudinal cross-baffle, are very thin, flexible strips of metal clamped to both long sides of the baffle, as shown in Figs. 28.4 and 28.5.

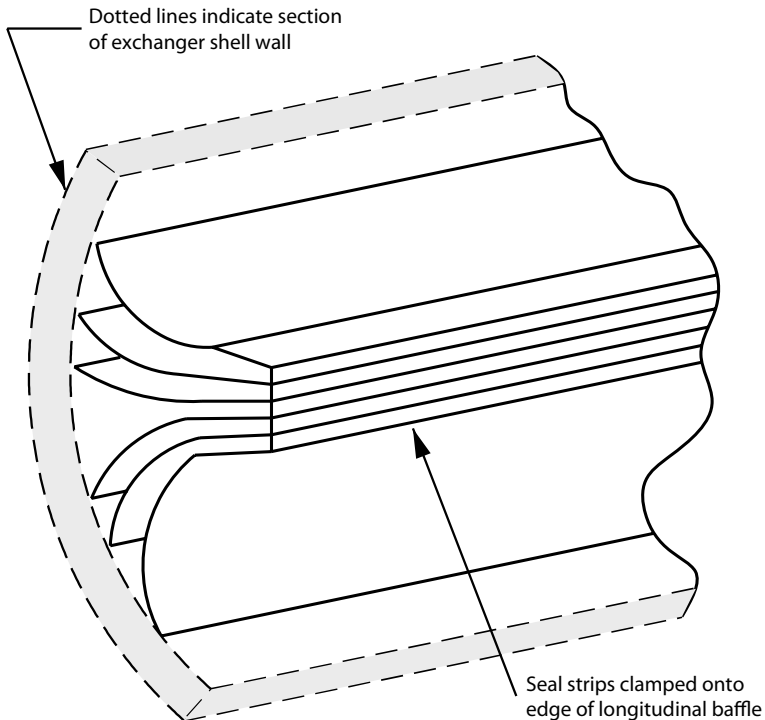


FIGURE 28.4 Detail showing portion of longitudinal cross baffle seal strips.

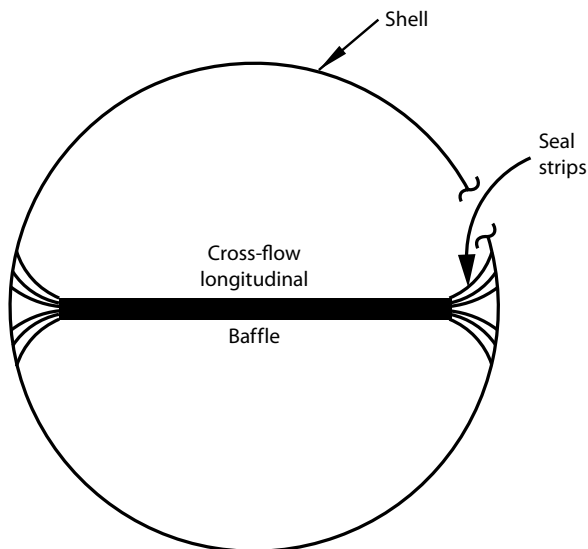


FIGURE 28.5 Longitudinal baffle.

There are two problems with these strips. First, they are subject to corrosion unless the proper material has been selected. Second, they must be replaced each time the bundle is pulled.

If these conditions are not met, then the shell-side liquid will partially bypass the majority of the tube surface area, and heat-transfer rates will suffer.

I have seen new exchangers of “two-pass shell” configurations work well when new. However, with time and typically after a turn-around, performance has been degraded, due to leakage around the cross-flow (longitudinal) baffle.

My former employer, American Oil, prohibited their use because of this problem. I also adhere to this practice and thus will select placing two exchangers in series, which is the more expensive option, as opposed to relying on the two-pass shell alternate design.

For most applications, 316 s.s. is used for seal strips as flexibility and corrosion resistance are required. In noncorrosive services, copper alloy seal strips are used.

28.2.10 Selection of Fouling Factors

I see a lot of variation in the industry as to how designers specify exchanger fouling factors. As a conservative designer, I tend to use larger fouling factors than most. Much better to add in a safety factor this way than to assume larger than anticipated flows. If the flow is much less than the design, the velocities will also be too low.

A big fouling factor is typically 0.010 to 0.020 inch (hr) (ft²) (°F) ÷ Btu. For example, I might use 0.025 for extract from tar sands. A really nasty service. Or visbreaker fractionator bottoms. Or resid from a deep-cut vacuum tower.

For clean, light crude oil downstream of the desalter, I would use 0.003. For heavy Venezuelan crude upstream of the desalter, I would use 0.007 to 0.010.

For jet or diesel, I would use 0.002 to 0.004. For heavy gas oil, about 0.005.

For light hydrocarbon reboilers, about 0.003 would be okay for virgin services and 0.005 for thermally degraded products that have not been exposed to air. If you have a thermally degraded naphtha that has been exposed to air, then 0.010–0.020 would be a reasonable fouling factor.

The steam side of exchangers is not subject to fouling. Yet I would still use 0.001, just to be conservative.

For air coolers, I would also use an air-side fouling factor of 0.001. What is far more important on an air cooler is the tube-side fouling factor.

The external finned area of an air cooler is typically 20 times larger than the tube internal area. Thus, if a tube-side fouling factor of 0.002 is used, it must be multiplied by 20 to obtain the effective fouling factor, based on the air cooler finned area.

If the overall heat-transfer coefficient for the air cooler is 10 Btu/hr/ft²/°F, then the 0.002 tube-side fouling factor will represent 24 percent of the total heat-transfer resistance:

- Step One: $0.002 \times 12 = 0.024$ where 12 is the ratio of internal area to external finned area.
- Step Two: The reciprocal of the heat-transfer coefficient (10) is the sum of the individual heat-transfer resistance 5 (i.e., $1/10 = 0.10$).
- Step Three: $0.024 \div 0.10 = 24\%$.

For ordinary tubes, I never bother about the small effect of the ratio of outside area ÷ inside area. But for finned, studded, or serrated tubes (i.e., low fins), this ratio of area factor is really important, as it magnifies the effect on the tube-side fouling factor that is selected.

28.2.11 Selection of Allowable ΔP

This is a complex and difficult subject. The larger the specified design ΔP , the more latitude the exchanger designer has to configure the exchanger for high velocities and a lower Reynolds number.

If we specify too low a ΔP , then we force the exchanger designer to specify low velocities.

If we specify too high a ΔP , then the designer may not choose to consume the available ΔP , which is then wasted across a control valve.

So, I guess my approach is best. Specify yourself:

- Size, length, and number of tubes
- Tube ID and pitch
- Number of tube passes
- Baffle spacing on shell side
- Baffle cut

Then, you will get the exchangers you really want every time.

CHAPTER 29

Fired Heaters: Fire- and Flue-Gas Side

Draft and Afterburn; Optimizing Excess Air

Next to the efficiency of the trays in a distillation tower, the efficiency of a fired heater is the most critical factor in saving and/or making money for the process plant. The primary objectives in operation of a fired heater are to

- Keep the fire in the firebox
- Avoid excessive heat density in the firebox
- Maximize the process heat absorption for a given amount of fuel

These objectives are equally important in the operation of fired boilers whose principles of operation on the fire side and flue-gas side are essentially the same as those of process-plant-fired heaters.

The two governing modes of heat transfer in the construction of a fired heater are radiation and convection.

In the firebox, heat transfer by radiation reigns supreme governed by Lambert's laws, as follows:

$$Q_R = A \cdot \varepsilon (T_{fb}^4 - T_{tm}^4)$$

where Q_R = radiant heat-transfer rate, Btu/h

A = surface area of the radiant tubes in the firebox, ft²

ε = emissivity factor

T_{fb} = absolute temperature, °Rankin, of the radiant surface,
which is essentially the firebox temperature. The reason

for this is that the flames heat not so much the tubes as the refractory, and the refractory then reradiates the heat to the tubes, so the main heat source becomes the refractory

T_{tm} = the receiving metal absolute temperature, °Rankin (this is the radiant tube metal or tube skin temperature).
(Rankin = °F + 460)

The amount of heat transferred to the process fluid in the convective section of a heater is governed by

$$Q_c = A \cdot U(T_{fg} - T_{bulk})$$

where Q_c = convective heat-transfer rate, Btu/h

A = surface area of the tubes, ft²

U = a heat-transfer coefficient, Btu/[(h)(ft²)(°F)]

T_{fg} = temperature of the flue gas, °F

T_{bulk} = temperature of the process fluid flowing inside the tubes, °F

Note that the temperature driving force for radiant heat transfer ($T_{fb}^4 - T_{tm}^4$) is always a very large number as compared to the temperature driving force for convective heat transfer ($T_{fg} - T_{bulk}$). For this reason, in nonfouling services we use finned tubes in the convective section of our fired heaters to increase the surface area of the tubes. The tubes in the firebox or radiant section are made of high-chrome steel capable of withstanding firebox temperatures of up to 2000°F depending on the severity of operation for which the heater has been designed. (Some very high-severity heaters may be designed for firebox temperatures of >2000°F.) Note especially that radiant section tubes are protected from overheating by the cooling effect of the process fluid flowing inside them, where “flowing” is the key word. The bare, unfinned, shock tubes are also cooled by the process fluid flow (Fig. 29.1).

However, the convective-section finned tubes are not intended or designed to withstand such high temperatures, and so the tubes themselves are often made from low-temperature-rated carbon steel. The fins, which are not cooled by the process flow, are made from chrome steel. As it is much easier to make finned tubes from just one type of metal instead of two, though, the furnace manufacturers will often choose to make the finned convective-section tubes entirely out of low-chrome steel, in which case one could expect them to withstand temperatures of ≤1300°F. It is advisable to check what types of tubes you have in your furnace and *know* what temperature tolerances they have. The only thing that prevents the tubes in the convective section from overheating up to firebox temperatures is that we keep the fire in the firebox and do not allow the fire to get up into the convective section. This may sound obvious, but it is surprising how many people seem to forget or ignore this fact.

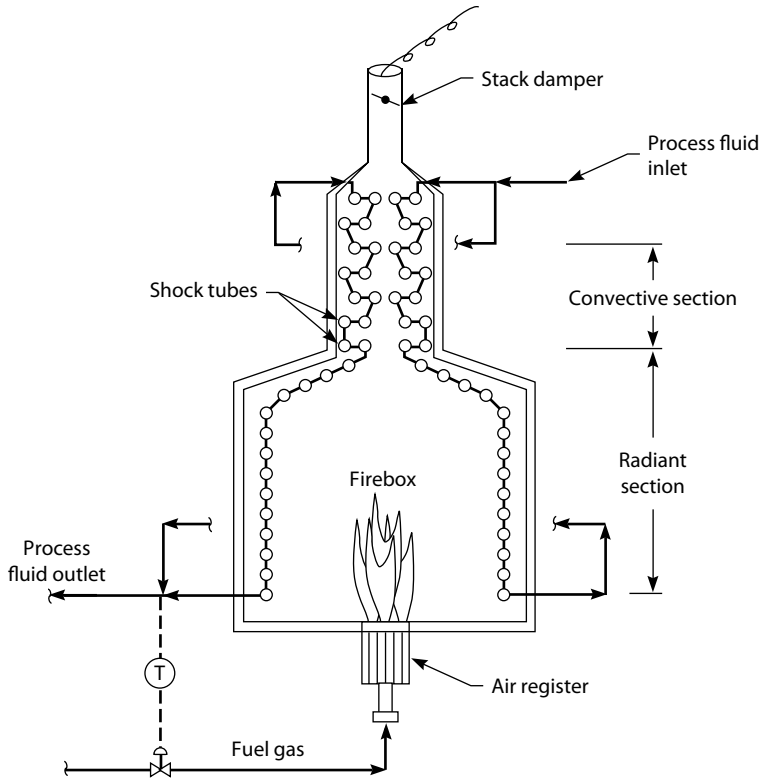


FIGURE 29.1 Typical natural-draft gas-fired process heater.

29.1 Effect of Reduced Air Flow

A typical natural-draft gas-fired process heater is shown in Fig. 29.1. Suppose we gradually close either the stack damper or the air register; the flow of air into the firebox will then be reduced. If both the process-side flow and the fuel-gas rate are held constant, the following sequence of events occurs:

1. The heater process outlet temperature begins to increase as the excess air is reduced. This is because more heat is given to the process fluid and less heat goes up the stack.
2. The heater process outlet temperature declines as air flow is reduced past the *point of absolute combustion* as defined in the next section. In this situation we have products of incomplete or partial combustion such as aldehydes, ketones, and carbon monoxide going up the stack. This sets the heater up for *afterburn* in the stack, and the heating value of the fuel is also effectively reduced.

Allowing a fired heater, boiler, or furnace to operate with insufficient air is hazardous because

- The products of incomplete combustion are hot and will ignite as soon as they find sufficient oxygen. This usually results in afterburn in the convective section or stack and can even lead to explosive detonations.
- The products of partial combustion are atmospheric pollutants.
- It is not possible to operate in automatic temperature control on the wrong side of the point of absolute combustion.

29.2 Absolute Combustion

Consider a forced-draft boiler producing 600-psig steam as shown in Fig. 29.2. The fuel rate on this boiler is fixed and we are going to optimize the oxygen (O_2) content of the flue gas by adjusting the speed of the forced-draft (FD) fan. Do we simply adjust the FD fan to give 2 percent O_2 in the stack because someone once said that 2 percent O_2 in the stack was a good number?

No! We are going to adjust the speed of the FD fan to produce the maximum amount of 600-psig steam. In other words, we are going to

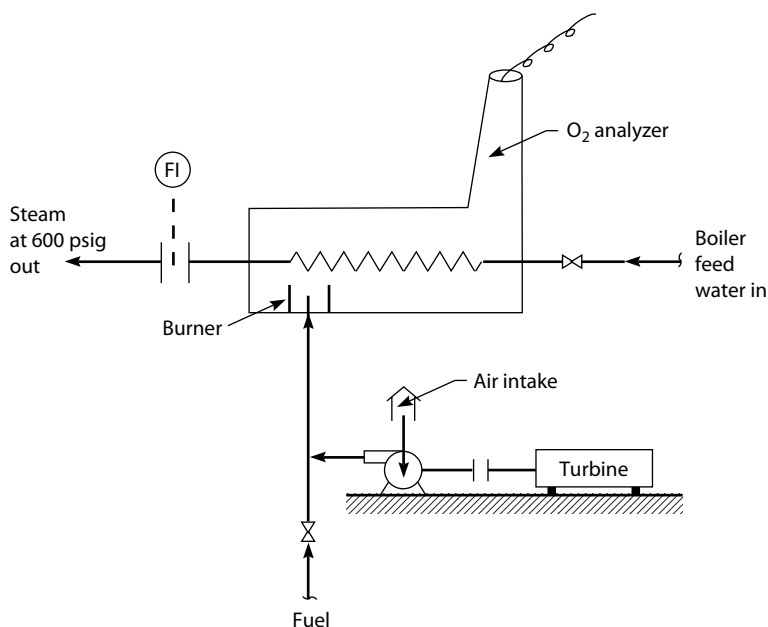


FIGURE 29.2 Forced draft-fired boiler; simplified schematic drawing.

maximize the heat to the process by adjusting the air rate. The point at which the steam production is at maximum is the point of absolute combustion. The oxygen content of the flue gas at the point of absolute combustion, where steam production is maximum for a given amount of fuel, will be the optimum; it will not necessarily be at 2 percent O_2 or any other fixed oxygen content. The oxygen content of the flue gas at the point of absolute combustion is a variable, depending on air-fuel mixing efficiency.

The term *absolute combustion* is not the same as complete combustion. *Complete combustion* is a theoretical term, implying a theoretical goal that we might aim toward but will never quite reach on any real process heater or boiler. The point of absolute combustion represents the best achievable efficiency point of any such piece of equipment on any day of the week, at any hour or minute.

One definition of the point of absolute combustion is the point of maximum heater outlet temperature for a given amount of fuel (as illustrated in Fig. 29.3). Following this, we can say that the point of absolute combustion is also the point of best or maximum achievable combustion for a given amount of fuel (as illustrated by Fig. 29.4). The point of absolute combustion in terms of either maximum heater outlet temperature or minimum fuel fired for a given outlet temperature would correspond to the same flue-gas oxygen content for a given furnace at the same moment in time. However, we must also note that this flue-gas oxygen content that corresponded to the point of absolute combustion on, say, heater 3304-A on Tuesday, January 2, 2006, at midday is not likely to correspond to the point of absolute combustion on the same heater on Wednesday, January 3, 2006, at midday,

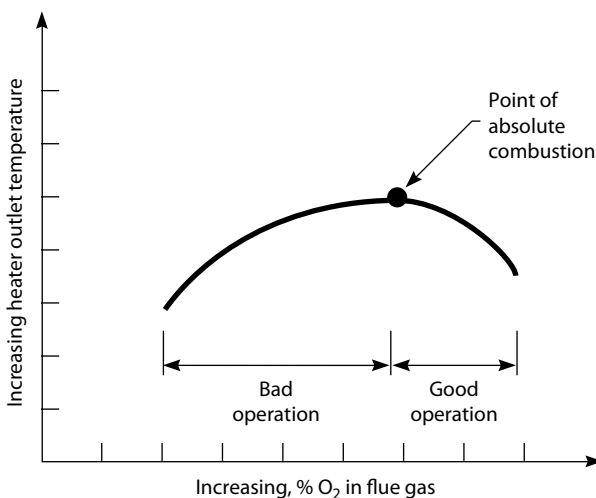


FIGURE 29.3 Point of absolute combustion in terms of heater outlet temperature.

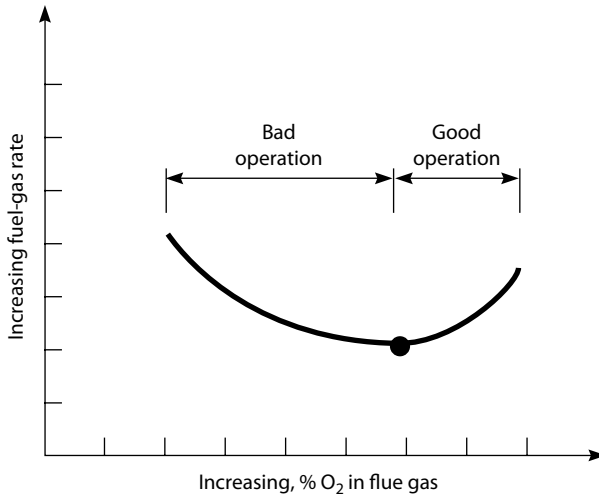


FIGURE 29.4 Point of absolute combustion in terms of maximum achievable combustion of fuel.

or even Tuesday, January 2, 1996, one hour or one minute after it was first determined. Indeed, the oxygen content (percent O₂), of the flue gas typically varies in the range of 0.5 percent O₂ to as much as 10 percent O₂ depending on the design, mechanical integrity, and operating characteristics of the equipment, such as firing rate, burner type, and ambient weather conditions.

Did the oxygen analyzer really help us find the point of absolute combustion for the forced-draft boiler? No, it did not. However, once you have found the point of absolute combustion and then noted the corresponding flue-gas percent O₂, as long as all operating conditions remain constant for the heater, then you could use the percent O₂ in the flue gas as a rather secondary guideline for that period of time until the operating conditions changed.

29.2.1 Oxygen Starvation

If you try to operate a fired heater with too little combustion air you will starve the burners of oxygen to “smother” or “bog down” the fire-box, then you will likely cause “afterburn” or secondary combustion in the stack. You will not be able to operate on automatic temperature control, and you may even destroy the equipment altogether.

This type of operation can also be described as operating on the wrong side of the point of absolute combustion. Take another look at Figs. 29.3 and 29.4; we are talking here about the portion of the curves marked as “bad operation.” What would be the color of the flue gas as it emerged from the stack during this type of operation? Would it be black?

No, it would not necessarily be black.

If we were burning heavy industrial fuel oil with a very high carbon:hydrogen ratio, then yes, very soon after we began to operate on the wrong side of the point of absolute combustion, the stack gases would turn black. After all, the black color is merely soot or unburned carbon. However, if we were burning pure hydrogen, the stack gases would never turn black no matter how much we starved the firebox of oxygen. If we were burning natural gas, then as we crossed over the wrong side of the point of absolute combustion, we might initially just be able to see the stack gases turn only a pale yellow. If we continued to decrease the combustion-air rate, the stack gas color might progress from pale yellow to dark yellow, to light orange, to dark orange, to brown, and finally to black. The color of the stack gases is not an accurate indication of whether we are using enough combustion air, as the point at which the stack gases finally turn black depends on the carbon:hydrogen ratio of the fuel. Also, while continuing to fall below the point of absolute combustion, we accelerate the production of atmospheric pollutants, such as aldehydes, ketones, light alcohols, carbon monoxide, and other products of the partial combustion of light hydrocarbons in the stack gases.

If we have the furnace on automatic temperature control while we are not using enough combustion air, and if the control valve on the fuel gas then opens to allow more fuel to the burners in order to increase the furnace outlet temperature, the extra fuel will not burn efficiently. In fact, the extra fuel is likely to *reduce* the heater outlet temperature rather than increase it, as there is already a shortage of air and it cannot burn properly and tends to cool the firebox. The automatic temperature controller then senses the reduction in heater outlet temperature and increases the fuel rate. Thus the furnace will spiral into an increasingly dangerous condition as the outlet temperature continues to fall and the furnace is left on automatic control. The way out of this situation is to put the furnace on manual control and manually reduce the fuel gas back to the point of absolute combustion.

To determine if you are running above the point of absolute combustion (i.e., the safe area), observe the heater outlet temperature as you manually reduce the fuel-gas rate. If this temperature increases, then you are still on the wrong side of the point of absolute combustion. If this temperature decreases, then you are on the safe side of the point of absolute combustion. Do not open the burner air registers until you are on this safe side. Increasing air flow when operating in a fuel-rich environment may make the firebox explode.

However, if we operate on manual without enough combustion air, we will share the experience explained by one oil refinery operator, as follows:

Our crude unit heater is very old; we have to operate it manually. We have three air blowers to produce the combustion air; we always run two with one on

standby. Once a week we switch the blowers by shutting one down and turning on the one that was idle.

One week I was switching the blowers when I forgot to shut one of the two down before starting up the third blower. I then had all three blowers running at once. Our normal heater outlet temperature target is 680°F, but while I had all three blowers on, the outlet temperature rose rapidly to 740°F.

Rather than cutting the fuel gas back to reach a 680°F outlet temperature increasing excess oxygen, he shut down the third blower and dropped the furnace temperature back to 680°F but also continued to run fuel-rich, without enough air.

Which is cheaper, air or fuel? Of course, the answer is *air*. Are we trying to make the flue gas to some particular oxygen-content specification, or are we trying to save fuel? Certainly, the answer is that we want to get the most heat we can out of every pound of fuel. Well, then, dear reader, let us neither set nor encourage the use of these stack or flue-gas oxygen targets; they are meaningless and often cause more harm than good.

29.2.2 Appearance of the Firebox and Flames

Regardless of and in preference to oxygen analyzer results, we must consider the appearance of the firebox and flames when we assess whether more combustion air is needed:

If the firebox appears bright and clear, then there is more than enough oxygen (a crystal-clear firebox has a large excess of oxygen).

If the firebox appears hazy and the flames are long, licking, yellow, and smoky-looking, then there is not enough oxygen.

To be just right, the firebox should have a very slight haze and all the flames should be compact and not searching around the firebox looking for oxygen.

Do not stand directly in front of an inspection port while you open it, in case there is positive pressure behind it.

Flame color depends on fuel composition. Gas often burns blue, but heavy fuel oil burns yellow. A yellow flame is caused by thermal cracking of the fuel. There is nothing wrong with a yellow flame; it is the general shape of the flame that is important. Flames should be short and bright.

The temperature of the inside of a firebox can also be estimated by its visual appearance (Table 29.1).

29.2.3 Secondary Combustion or Afterburn

In the preceding paragraphs we discussed how operation without enough combustion air on the wrong side of the point of absolute combustion leads to inefficient combustion of fuel. Apart from the fuel wastage, there is another important disturbing and frequently

Color	Temperature, °F
Dark blood red or black red	990
Dark red	1050
Dark cherry	1250
Bright cherry	1375
Light red	1550
Orange	1650
Light orange	1725
Yellow	1825
Light yellow	1975
White	2200
Dazzling white	2730

TABLE 29.1 Visual Estimation of Temperatures

occurring problem associated with this bad mode of operation. The products of incomplete combustion of the fuel, and in more severe cases even the unburned fuel itself, flow with the flue gas up through the convective section and up the stack. A certain amount of air leaks in through the convective section and stack from outside the heater. When these hot, combustible hydrocarbons in the form of unburned fuel mix with the extra air that is leaking in from the environment, re-ignition is liable to occur. We call this *afterburn*, or *secondary ignition*, meaning there is now fire in the convective section.

When combustible materials or unburned fuel re-ignite in the convective section, a dramatic increase in flue-gas temperature will occur. The metallurgy of finned carbon steel convective-section tubes is not designed to withstand high temperatures. The fins will become oxidized and, when cool, may become brittle and thicker than before, thus restricting flue-gas flow. The convective-section tubes themselves will become warped and bent and thus restrict flue-gas flow still further.

Just such a thing happened as a result of another well-intentioned but misguided attempt to improve fuel firing efficiency by reducing the oxygen content of flue gas in a fired heater. This heater, which was also manually operated, had an oxygen analyzer probe installed in the convective section. The operators were given a target range to run between 2 and 6 percent O_2 (note that this range would in many cases actually include the point of absolute combustion).

The inside operator (operator A), noting that the stack gases had 8 percent O_2 , began to cut back on the combustion air. As operator A pinched back on the air register, the convective-section oxygen dropped slowly from 8 to 7.5 percent O_2 . The outside operator (operator B) went to check on the firebox and, seeing that it was full of

long, licking, yellow, smoky flames and had taken on a very hazy appearance, reported back to operator A that he considered the amount of air in the firebox was insufficient.

Meanwhile operator A looked again at his O_2 analyzer output, which showed 7.5 percent O_2 . He told operator B that there was plenty of oxygen because the analyzer showed 7.5 percent O_2 , so he must be mistaken and, in fact, that he needed to go and pinch back still further on the air register to cut back the combustion air.

After cutting back on the air registers, operator A noticed that the heater outlet temperature dropped, so he raised the fuel rate to maintain the heater outlet temperature. He did not realize that this sequence of events was an indication of insufficient combustion air, because the convective-section oxygen analyzer still showed 7 percent O_2 , so he asked operator B to pinch back still further on the air registers.

At this point, black smoke started coming out of the stack, and several of operator B's colleagues began to suggest that they should start to open up the air registers and increase the combustion air and not reduce it any further. However, operator A, now reading 6.5 percent O_2 in the convective section on his O_2 analyzer, was determined to reach his 6 percent O_2 , and requested a further cut-back on the combustion air.

When the combustion air was cut back this time, fire started to come out of the stack. Now operator A was forced to admit that they would have to increase the combustion air again. The problem they then faced was that the amount of draft in the heater seemed to be less than before and they were not quite able to re-establish the same air flow. The reason for the restricted air flow was that they had caused afterburn in the convective section and damaged the convective-section tubes, which now restricted the flow of flue gas and hence the flow of combustion air to the burners.

29.2.4 Using the Concept of Absolute Combustion in Operations

The only correct oxygen target is that firebox oxygen content that maximizes the process-side heat absorption for a given amount of fuel, or if you prefer, we could say it is that the firebox oxygen target that minimizes fuel consumption for a given process-side heat absorption.

This correct percent O_2 in the firebox can be determined only by experimenting with the fired heater, furnace, or boiler in the field. This firebox O_2 will also vary for any given piece of fired equipment depending on a variety of factors, including mechanical integrity and firing rate. In reality, it is not the firebox oxygen content that we are looking for so much as the correct combustion-air rate for proper operation. The search for the correct combustion-air rate on any particular day is known as *heat-proving the heater*. Experienced operators will increase and decrease the combustion air to find that particular rate

that minimizes the fuel rate. Naturally, experienced operators have been heat-proving heaters since long before oxygen analyzers were invented. In view of this, it seems that we do not actually need an oxygen analyzer to run a piece of fired equipment efficiently.

The concept is not to run the fired equipment to a specific oxygen target because the oxygen content corresponding to the most efficient operation varies, but to aim instead to maximize heat absorption to the process stream for a given fuel rate. Once we have heat-proved the heater and found this point of maximum heat absorption, which we call the *point of absolute combustion*, we will then need to operate the furnace with somewhat more air. We call this operating on the “good” side of the point of absolute combustion, as shown in Figs. 29.3 and 29.4.

If we do not ensure that we operate some extra air to put us on the good side of the point of absolute combustion, then we run the risk of getting into oxygen starvation on the wrong side or “bad” side of the point of absolute combustion.

Automatic operation linked to process outlet temperature while on the bad side of absolute combustion is potentially hazardous because the heater outlet temperature will drop as a result of the reduced heating efficiency of the fuel. The automatic control will then call for more fuel, which, in turn, produces more oxygen starvation. The only way out of this situation is to put the furnace back to manual operation and manually reduce the fuel to get back to the good side of the point of absolute combustion.

Note that combustion air is inexpensive compared to fuel; it is irrational to cut back on combustion air and waste fuel by inefficient burning. Also note that to operate fired equipment efficiently, we need to heat-prove it or test to find the point of absolute combustion on a regular basis.

29.2.5 Flue-Gas Oxygen and Tramp Air

Suppose we have a natural-draft heater operating very efficiently on the good side of the point of absolute combustion. The oxygen content of the firebox gases (just below the shock tubes) is 2.5 percent oxygen as shown in Fig. 29.5. What do you think the oxygen content of the flue gases in the stack will be?

The answer is it will *always* be more than the firebox oxygen content, because there are *always* air leaks in the convective section and stack that, with the slightest negative pressure in the stack or convective section, will allow air to leak in. These air leaks are collectively known as “tramp air.”

The amount of tramp air will vary according to

- The mechanical integrity of the heater, i.e., the number of holes in the casing
- The firing rate and draft balance of the heater

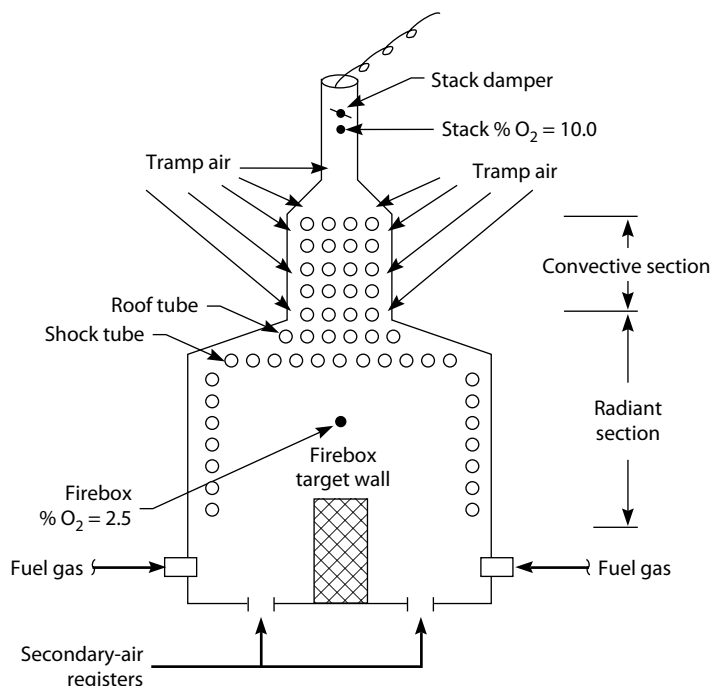


FIGURE 29.5 Natural-draft heater showing tramp air leaks.

The oxygen measured in the stack is the sum of the unused oxygen from the firebox plus the oxygen from tramp air drawn into the convective section and stack.

We have already shown that it is wrong to operate a furnace to an arbitrary oxygen target. It is entirely incorrect to adjust the furnace combustion air based on stack gas and/or convective-section flue-gas oxygen readings, because—as a result of the inevitable air in-leakage—the stack gas and/or convective-section flue-gas oxygen content does not necessarily represent the amount of oxygen in the firebox. It is also impossible to offset the stack or convective-section oxygen readings to represent the firebox oxygen content because the amount of air in-leakage varies so much with time as conditions change.

Unfortunately, many heaters have oxygen sensors in the stack and not in the firebox. An oxygen sensor in the firebox may provide a useful guide, as we mentioned earlier, although it is not actually needed to run the heater efficiently. However, an oxygen analyzer in the stack or convective section is really useful only in operations if there is also an oxygen analyzer in the firebox; in such instances it could be used to trend air leaks across the convective section. Never ever use an on-line analyzer for technical analysis or test work. Oxygen-content field measurements should always be taken with a single portable oxygen

analyzer. Be careful to calibrate the portable instrument for the range in which we want to use it. Cylinders of test gases with specific oxygen contents are available for calibration purposes.

If the combustion of air flow is being adjusted on the basis of the oxygen content of the stack or convective section gases (i.e., from readings taken on an oxygen analyzer with a probe located in the stack or convective section), there may be afterburn or secondary ignition in the convective section. The fire is supposed to be contained within the firebox, and not in the convective section. If we intend to install an oxygen analyzer, the analyzer should be monitoring oxygen in the firebox itself. Lone oxygen analyzers monitoring oxygen in the convective section are misleading as a result of the tramp air, and are a positive hazard to safe operation.

29.3 Draft

29.3.1 Draft Readings

Draft readings are a comparison of two pressures taken *at the same elevation* and are traditionally quoted in *inches of water*. To make sense of a set of draft readings for a furnace, you must first normalize the data.

In Fig. 29.6, we see a simple natural-draft heater with no convective-section tubes. The laws of hydraulics tell us that fluids flow from regions of high pressure to regions of lower pressure, and yet the draft readings in Fig. 29.6 seem to contradict this principle.

Notice that the draft readings are made at different elevations. Each measurement is in reality a comparison between the densities of the gas both inside and outside the furnace at a given elevation. The temperature difference is the main reason for difference in density inside and outside the furnace. The molecular weights of the furnace gases and of air are approximately the same.

To make sense of the draft measurements so that we can use them to evaluate furnace pressure drop, proceed as follows:

1. Make a datum line across the top of the stack as shown in Fig. 29.6.
2. For each draft reading, *add* on the pressure exerted by the appropriate static head of air.

Given: 150 ft of air = 2.22 inch H_2O (or inches of water gauge), in our example, which is for 60°F ambient air at sea level. Figure 29.6 illustrates the principle.

The standard instrument which is used to take draft readings is the magnehelic delta-pressure (ΔP) gauge. However, if you wish to carry out a draft survey, it is possible to make some fairly good readings with nothing more complicated than a clear glass bottle, some water, and a length of clear plastic tubing. Fill the bottle with water,

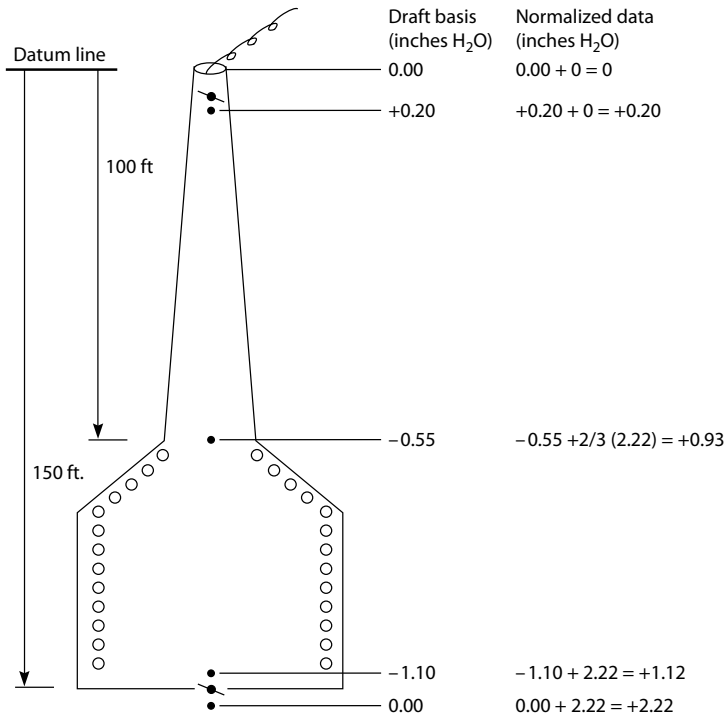


FIGURE 29.6 A simple natural-draft heater with no convective section tubes.

insert the plastic tube in the bottle. Attach the other open end of the plastic tube at the point where you wish to take the draft reading. The difference in the vertical height between the level of water in the bottle and the level of water drawn up in the tube will be your draft reading in inches of water gauge, as shown in Fig. 29.7.

29.3.1.1 Draft Induced by Wind

Running with a positive pressure below the bottom row of convective tubes is potentially dangerous, especially if there is an appreciable quantity of sulfur in the fuel. The sulfur dioxide would then blow out of the sight ports and leak in the convective section. SO₂ is a toxic gas.

In some texts, it is advised to maintain a minimum safe draft of 0.05 inch H₂O. However, there is a problem with this target. The problem is wind. Wind can create a variable draft of plus or minus 0.10 inch H₂O or more. If the draft decreases, as the gusts of wind come and go, then the heater may go positive and a personnel hazard is created. Thus, the minimum safe draft depends upon the variability of the wind. Not at grade, but the wind speed at the top of the stack.

When draft decreases, due to a reduction in wind, air flow into the burners also decreases. If this causes the firebox to drop below the point

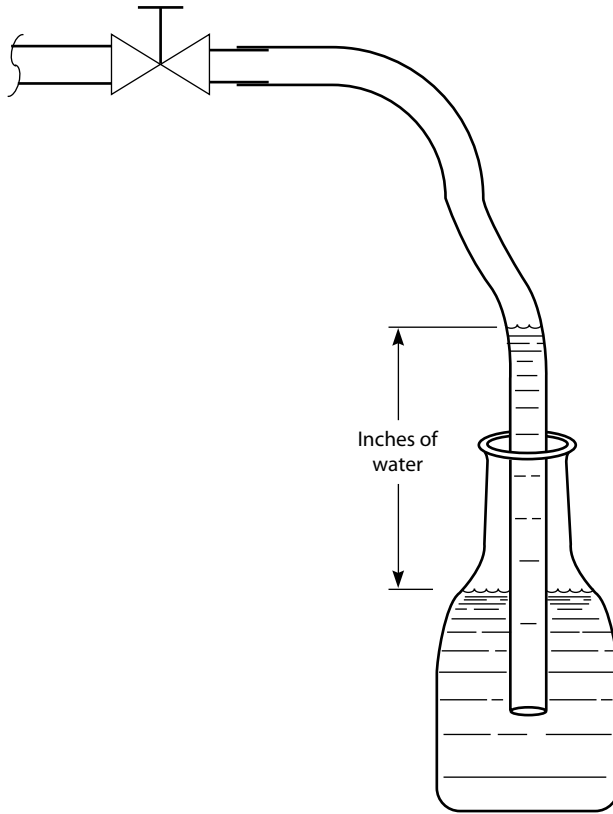


FIGURE 29.7 A clear bottle and a flexible tube can be used to make approximate draft measurements.

of absolute combustion, then the heater will bog down and become flooded with fuel. That is, the heater will be caught in a positive feed-back loop, on the wrong side of the point of absolute combustion.

29.3.1.2 Draft Induced by Density Difference

Draft is mainly due to the density difference between the hot flue gas and the ambient air. Draft is also a function of the height of the stack. For example:

- Density of cold air = 0.070 lb/ft^3
- Density of hot flue gas = 0.030 lb/ft^3
- Density difference = 0.040 lb/ft^3
- Stack height = 100 feet

The calculated draft is then:

- $0.040 \text{ lb/ft}^3 \times 100 \text{ ft} = 4.0 \text{ lb/ft}^2$

To convert to inches of H_2O :

- $4.0 \text{ lb/ft}^2 \div 144 \text{ (inch)}^2/\text{ft}^2 \times 28 \text{ inch } H_2O/\text{PSI}$

Therefore, Draft = 0.80 inch H_2O .

The wind also creates 0.1 to 0.2 inches of water draft. Frictional losses in the stack tend to reduce draft. I usually ignore both factors, even though on occasion they may very well be significant.

29.3.1.3 Draft Induced by Reduction in Stack Diameter

We were driving through some very rural and remote areas of India last year. The major crop was sugarcane. Every few miles there was a primitive process plant to produce raw sugar from the cane. The cane was crushed by electric motor-driven rollers to recover the sweet juice. The juice was then boiled down and partially evaporated, using the crushed cane as fuel. I noted that the top of the boiler chimney was about one-third of the diameter at the base of the chimney. The reason for this large reduction in chimney diameter is to help induce a draft by increasing velocity of the flue gas in the upper portions of the chimney. Neglecting frictional losses, the draft induced by increasing velocity due to the reduction in chimney cross-sectional area is (see Section 16.4.2 on “Nozzle Exit Loss Calculation” in Chap. 16 for details):

- $\text{Draft} = [(\text{Density})/62.3] \cdot (0.186) \cdot (V_2^2 - V_1^2)$

where:

- Density = Density of the flue gas in lb/ft^3
- 62.3 = The density of water, lb/ft^3
- 0.186 = The conversion factor used to convert kinetic energy into potential energy
- V_1 = Velocity, ft/s , in fat part of chimney
- V_2 = Velocity, ft/s , in thin part of chimney

(Note—The 0.186 coefficient is based on the gravitational constant, 32.2 ft/s^2 .)

If the diameter of the chimney is reduced by two-thirds, then the chimney cross-sectional area is reduced by 90 percent. Hence, if the velocity at the bottom of the chimney (V_1) is 10 ft/s , then the velocity at the top of the chimney (V_2) is 100 ft/s .

Therefore, the draft due to the reduction in chimney diameter is:

- $(0.035)/62.3 \cdot (0.186) \cdot (10,000 - 100) = 1.05 \text{ inches of water draft}$

The 0.035 is the density of the flue gas at 600°F , assuming its molecular weight is the same as air.

Caution: Based on the roughness inside the stack, some or all of this draft may be lost due to friction.

We asked the local guys running the boiler why they thought the top portion of their stack was so narrow. They simply answered, "This is the way it's always done, it is our tradition handed down for generations. This is India, and here we have learned that it's best to follow our traditions."

29.3.2 Draft Balancing

The air register and the stack damper are used together as a team to optimize the heater draft. Our aim in balancing the draft for a natural-draft or a balanced-draft heater is to maintain a small negative pressure of -0.1 inch of water, just below the shock tubes (refer to Fig. 29.5), at the entrance to the convective section. At the same time we must maintain enough air to operate on the good side of the point of absolute combustion.

If we operate with a positive pressure in the firebox, although the burners may appear to operate normally, the hot flue gases will leak outward. This damages the roof arch supports and the steel structure so as to shorten the life of the heater. Flame impingement of the upper radiant tubes may also result from positive pressures.

As we close the stack damper, the pressure at the convective-section inlet will increase; that is, we will have less draft. This will reduce the rate of flue gas to the convective section, minimizing the rate of air in-leakage to the convective section and the chances of afterburn occurring. However, if we close the stack damper too much, a positive pressure will develop at the convective-section inlet, which we must avoid.

Conversely, as we open the stack damper, the pressure at the convective-section inlet will decrease, so that we have more draft. If we have too much draft, we will increase the risk of afterburn as we are increasing the rate of air in-leakage in the convective section and stack.

We adjust the draft with the stack damper and maintain the combustion-air level by adjusting the air registers to accommodate the adjustments made on the stack damper.

29.4 Air Leakage

29.4.1 Evaluating Fuel Wastage Due to Air Leaks

Convective-section air leaks reduce the thermal efficiency of the heater by mixing cold ambient air with hot flue gas. The resulting energy debit may be expressed as follows:

$$\Delta F = \frac{(T_s - T_a)(O_{2,s} - O_{2,c})}{500}$$

where ΔF = percent heater fuel wasted through lost-heat recovery in the convective section

T_s = stack temperature, °F

T_a = ambient temperature, °F

$O_{2,s}$ = percent oxygen O_2 in stack

$O_{2,c}$ = percent oxygen in combustion zone, i.e., just below the shock tubes at the inlet to the convective section

Just for practice, let us suppose we have a stack temperature of 600°F, ambient air is 100°F, the convective section has 10 percent O_2 , and the firebox has 6 percent O_2 measured just below the shock tubes. What percentage of fuel fired (ΔF in our equation) is being wasted by cold air leaking into the convective section?

$$\text{Answer: } \Delta F = \frac{(10 - 6) \times (600 - 100)}{500} = 4\%$$

29.4.2 Minimizing Fuel Wastage Due to Air Leaks

Suppose we now cut back on the combustion air by closing the air registers a little. The oxygen content in the firebox is now 3 percent. The oxygen in the convective section will also be reduced to, say, 9 percent oxygen. Notice how the difference between the firebox oxygen and convective-section oxygen (which we refer to as DO_2) has increased. It is now 6 percent O_2 . This is because the draft through the heater has increased (i.e., the pressure is more negative), and more air is being sucked in through the holes or air leaks in the convective section. If we still had a stack temperature of 600°F with ambient air at 100°F, we would now be wasting 6 percent of the fuel fired. This is not a pleasing result, as we can see extra fuel will have to be burned in the firebox to offset the increased air in-leakage.

Suppose we put the air registers back as they were and pinch on the stack damper instead. So let's suppose we restrict air flow via the stack damper until the firebox oxygen goes down from our base case value of 6 to 3 percent. The oxygen in the convective section will also be reduced to, say, 5 percent. In this case, we see that the DO_2 is also much less, only 2 percent. This is because we now have less draft through the heater, that is, the pressure in the convective section increased and so has reduced the air in-leakage rate. This also illustrates how the air leaks themselves vary in accordance with the combined operation of a stack damper and air register.

29.4.3 Patching Air Leaks

To suppress afterburn and minimize energy losses caused by in-leakage of cold ambient air, any holes in the heater walls, convective section, ducts, and so on, should be patched. Also make sure that inspection ports are closed. Leaks can be detected on stream to a certain

extent by visual inspection (crumbling chalk dust or dropping a little baking powder past a suspected leak will pinpoint the leak).

During a turnaround, a smoke test should be carried out as follows:

1. Close the stack damper.
2. Ignite colored smoke bombs or even old tires in the firebox.
3. Turn on the forced-draft fan, if there is one, to assist the smoke.
4. Watch to see where the colored smoke emerges as these points are the sources of the air leaks.

Patch the leaks using heavy-duty aluminum tape, insulating mud, or silicone sealers, and weld up any loose sheet metal.

29.5 Efficient Air/Fuel Mixing

The function of a burner is to mix oxygen, in the form of air, with the fuel so that the fuel will burn more efficiently. Burners are available in a variety of different designs, all engineered with the intent to maximize air/fuel mixing efficiency. In more recent times we have the added concern of also minimizing the formation of atmospheric pollutants. Fuel will burn at the end of a pipe with no burner at all, but the combustion will be far from efficient.

Some burners are fitted with primary- and secondary-air registers, as is the premix burner shown in Fig. 29.8. Air entering through the

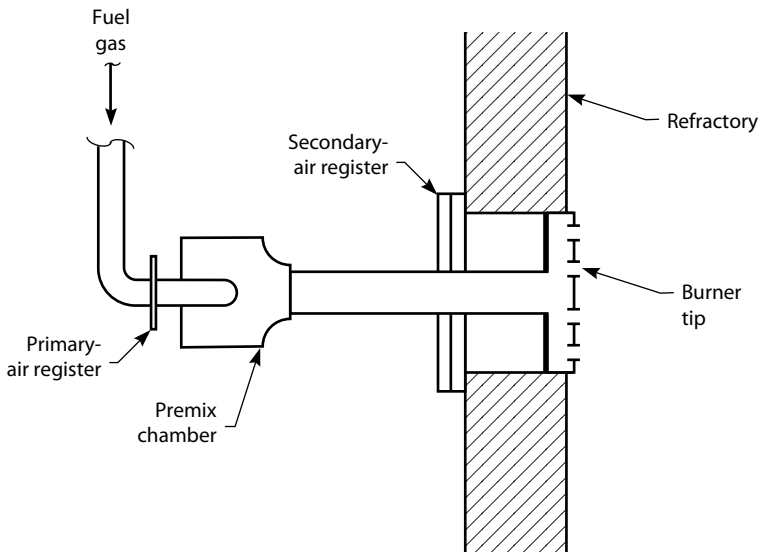


FIGURE 29.8 Schematic drawing of a premix burner showing primary- and secondary-air registers.

primary-air register mixes much more efficiently than does the air that enters through the secondary-air registers. Thus we should maximize the use of primary air, and we do this by gradually opening up the primary-air register until the flame just begins to lift away from the burner tip. The remaining combustion air should be provided through the secondary-air register.

29.6 Optimizing Excess Air

The term *optimizing excess air*, as we have seen, does not refer to operation at some arbitrary oxygen level; instead, consider the following in relation to your heater, and you will be optimizing excess air:

- Minimize fuel rate for a given heater outlet temperature (or steam production in a boiler), then operate at 0.5 to 1 percent higher oxygen.
- Maximize primary air to burners, where burners have primary and secondary air.
- Adjust draft to minimize air in-leakage while maintaining a small negative pressure at the entrance to the convective section.
- Close openings to pilot lights, sight-ports, and other holes around burners. (Combustion air only mixes properly through the burner air register.)
- When operating at reduced firing rates, shut down some burners if possible, as burners work more efficiently when operating close to their design capacity (also remember to close air registers on idle burners).
- Minimize poor lateral air distribution in the firebox by adjusting air registers on individual burners. Having a low air flow in one part of the heater will lead to higher overall oxygen requirements.
- Keep burners clean. Plugged burner tips increase oxygen requirements; maintain a regular burner cleaning program.
- Pay attention to the visual appearance of the firebox.

29.7 Air Preheating, Lighting Burners, and Heat Balancing

29.7.1 Air Preheaters

A typical air preheater will reduce the fuel required to liberate a given amount of heat by 10 percent. The debit for this improved thermal efficiency is a higher flame temperature and the possibility of overheating the radiant section. There is a clear advantage to fit an air preheater

to a furnace when the firebox is running below a maximum firebox temperature. Three types of air preheaters are in common use:

- Direct heat exchange between flue gas and air
- Heat exchange via intermediate circulating oil
- Heat exchange via massive heat-transfer wheel, packed with metal baskets (Lungstrom type)

The wheel type is subject to air leaks across the mechanical seals. This leakage is identified by increased oxygen content in the flue gas, low flue-gas outlet temperature, and a greater temperature loss in the flue than rise in the air temperature across the preheater, as shown in Fig. 29.9.

In general, air preheater leaks are a problem because they

- Reduce the thermal efficiency of the air preheaters.
- May reduce combustion air so that it falls below that required for absolute combustion, resulting in destructive afterburn. (*Note:* Air preheaters themselves are often destroyed by afterburn.)
- May cause the forced-draft fan to operate at higher flows, requiring higher driver horsepower.

All air preheaters are subject to corrosive attack caused by condensation of sulfur trioxide. At a 150-ppm sulfur in the fuel gas, operating experience shows that a minimum temperature of 350 to 400°F minimizes corrosive attack. Uneven cooling in the preheater results in the need to keep the flue-gas outlet temperature 50 to 100°F above the calculated SO_3 dew point. The flue-gas outlet temperature can be

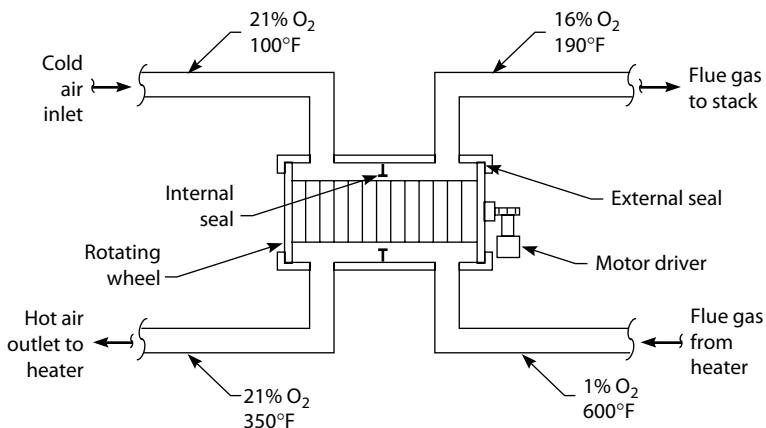


FIGURE 29.9 Rotating-wheel-type air preheater with leaking seals.

increased by use of an air preheater air-side bypass, or by increasing the excess air in the firebox (this second method is discussed further in Sec. 29.7.4).

29.7.2 Heater Thumping or Vibration

29.7.2.1 Preheater Vibration

Air preheaters or any type of waste-heat recovery device designed for horizontal flow across vertical tubes may be subject to vibration produced by the velocity of gas across the tube banks. The velocity produces a vortex-shedding wave pattern that could correspond to the natural harmonic frequency of the tube bank. If the natural harmonic frequency is reached, excessive vibration of the tubes will occur. Redesign of the internal baffle system by inserting dummy baffles can stop the vibration.

29.7.2.2 Vibrations in Fired Equipment and Afterburn

Balanced-draft or induced-draft furnaces and boilers are intended to be operated with a small negative pressure (ca. -0.1 inch H_2O) just below the first row of convective tubes, that is, just below the shock tubes. If we operate such a piece of equipment with a severe shortage of air in the firebox and massive air in-leakage in the convective section, it may sometimes experience thumping, puffing, or vibration.

Take a look at my sketch of a “puffing” boiler, Fig. 29.10. It is operating with perhaps a little too much draft, but look at the firebox oxygen; it is only 0.6 percent. However, the oxygen content of the flue gas steadily increases to 3.1 percent at the outlet from the convective section. This boiler has a tremendous amount of air leaking into the convective section. Perhaps someone is operating it on the basis of that 3.1 percent oxygen to the stack, without checking the firebox conditions—very foolish, as I hope you will now agree. This boiler is puffing, too; perhaps I should have shown its convective-section walls alternatively convex and concave, but instead I have just entitled it “the puffing boiler.” Here is what actually causes the puffing, vibrating, or thumping.

The fuel gas is not properly combusted at the burners, so partially combusted hydrocarbons and perhaps some totally unburned fuel pass up into the convective section with the flue gas. The convective section, as shown in Fig. 29.10, is under negative pressure, so air leaks in and mixes with the hot unburned hydrocarbons, which then re-ignite; thus we have afterburn in the convective section. The heat of the fire in the convective section expands in the flue gases, causing a positive pressure in the convective section, which stops the air leaks. After some period of time, the fire in the convective section has consumed all the tramp air and is extinguished. The convective-section flue gases then cool and shrink, re-establishing a negative pressure as before, and, as before, air leaks into the convective section

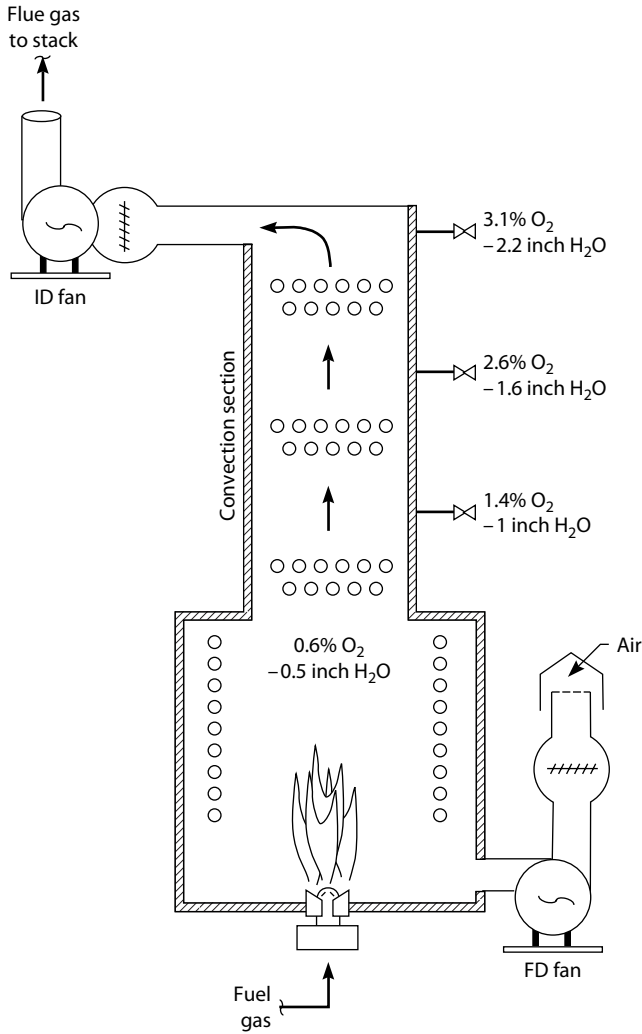


FIGURE 29.10 The “puffing” boiler: schematic drawing.

and the hot, unburned hydrocarbons from the firebox mix with that tramp air and re-ignite in the convective section.

This cycle of events will continue, except each time the ambient hydrocarbons ignite in the convective section and the pressure changes from negative to positive the leaks will increase in size as the boiler fabric is shaken.

If the air flow to the firebox becomes very severely low, then one or more of the burners may be extinguished through lack of oxygen. If the fuel is then left to flow unchecked, it will build up in the firebox

until eventually enough tramp air has been sucked into the firebox, at which point the extinguished burners may be re-ignited by the heat left in the refractory walls, causing a minor explosion or a detonation as it ignites. The magnitude of the explosion depends on the degree of oxygen starvation.

If anything like this is happening to your fired equipment, open up the air registers, get more air into the firebox, and patch the leaks!

29.7.3 Lighting Burners

Before lighting burners in a piece of fired equipment that has been idle or offline for any time or whose burners have become extinguished during operation, it is vital to first purge the firebox of hydrocarbons and combustibles or explosive mixtures. Here is a method as outlined to me by a prominent burner manufacturing company:

- 1a. If the furnace or boiler has a forced-draft fan, open the dampers and displace 5 volumes of air.
- 1b. If the furnace or boiler is natural-draft, open the air register and stack damper and wait for 5 to 10 minutes.
2. Check the firebox with a hydrocarbon detector (the plant's fire station will usually have one of these and someone trained in its use). Pay particular attention to stagnant areas of the firebox, as fuel-gas valves tend to leak through burners and gases build up in areas of low air flow. Look for unburned gas and explosive mixtures. If you detect any such fuel or explosive mixtures, repeat step 1a or 1b, then test again with the hydrocarbon detector. Continue this process until the firebox is purged of all combustible or explosive material.
3. Light the pilot lights.
4. Light the burners using pilot lights.

Please note that it is not only bad practice but extremely dangerous to light main burners without the use of a properly installed pilot light. Butane torches, "fireworks," torches consisting of gasoline-soaked rags on sticks, or residual heat in the furnace refractory *do not* provide a safe alternative to a properly installed pilot light. Please also note that fired equipment either without burner pilot lights or with extinguished burner pilot lights have been known to explode during operation, as the pilots, when lit, do provide some means of safely relighting or keeping main burners alight.

An explosive mixture in the firebox is indicated for methane (i.e., natural gas) when the methane content is between 5% and 15%. When the firebox methane content is below 5%, the air-fuel mixture is too lean to combust. When the methane content is above 15%, the mixture is too rich to combust. The comparable values for hydrogen are

4% (too lean) and 75% (too rich). This suggests that refinery fuel gas, which may contain 5 to 50% hydrogen, can explode much more readily than natural gas. The presence of H_2 in fuel gas is a strong argument, then, for connecting pilot lights to natural gas, which is very low in H_2 .

We have had several sad experiences of losing co-workers due to firebox explosions. Always, the deaths could have been avoided if the pilots had been lighted first.

For example, a survivor will say, "We tried to light the pilot lights first, but they wouldn't light-off. It isn't our fault that HDS heater blew itself up."

"Okay. It's not your fault. Some idiot connected the pilots to dirty fuel gas, rather than natural gas. But why didn't you clean out the little spud inside the pilot light? It only needs a crescent wrench to do the job in two minutes."

"That isn't our job," the operators replied.

"Okay. But if you can't get the pilot lit, then don't open the gas to the main burner."

"Well, how are we going to start up the HDS without H-3?"

"It's simple, guys. If you can't run a unit safely, then don't run the unit."

This was a pretty accurate rendition of a discussion with two operators at the Tenneco Oil Refinery in Chalmette, Louisiana, in 1984.

Today, burner manufacturers have access to all kinds of designs for pilot lights, to fit all shapes and sizes of burners and burner ports and draft systems from the smallest breeze to gale-force winds. There is no longer any excuse to operate without pilot lights installed and working.

29.7.3.1 Oil Burners

In the United States, oil firing is uncommon. In Europe, it's pretty standard. Most oil-fired burners use atomizing steam to produce a large surface area of the oil exposed to the combustion air.¹ In the steam atomizer shown in Fig. 29.11, jets of oil under pressure meet the spinning jets of steam under pressure in the mixing chamber. A mist of oil and steam is discharged into the furnace. The pressure drop of the oil through both the oil orifices of the tangential slots is typically 10 to 30 psi (0.5 to 2.0 BAR). In general, an oil-fired burner uses a higher percent of excess air, as it's more difficult to mix air with liquid fuel than with gaseous fuel.

The fuel oil flows from a circulating hot oil system kept at several hundred °F, depending on what is being burned. In most refineries, visbreaker or vacuum tower bottoms provide the fuel oil, which is circulated at about 500°F. Burner tip plugging of the oil orifices shown in Fig. 29.11 is a routine problem.

The most serious problem is loss of fuel oil circulation in case of an electrical power failure. The fuel oil can then solidify in the circulating

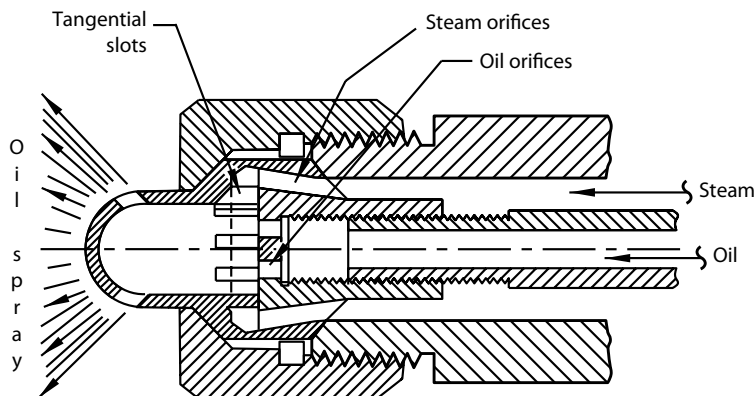


FIGURE 29.11 Steam-atomizing oil burner tip.

system and in the burners themselves. For example, vacuum tower visbreaker bottoms can solidify at 300°F.

29.7.4 Heat Balancing

Let us finally look at a situation where one might wish to transfer heat duty from the firebox to the radiant section. At the beginning of the discussion on air preheaters earlier in this chapter, we noted how fitting an air preheater to an existing furnace would increase the flame temperature and possibly overheat the firebox.

Suppose a fired heater in an oil refinery, in service to heat crude oil, is fitted with a new air preheater. The firebox was running hot before the air preheater was installed, but the management was convinced that the installation of the air preheater would allow them to heat more crude oil. When the heater with its new air preheater came back online, the following changes were noted:

- Fuel-gas consumption was down by 10 percent (for the same heat absorbed as before the revamp).
- The firebox, which was hot before the revamp, was now hotter, even at the original crude-oil flow rate; thus, rather than running more crude, operations personnel had to cut back the crude rate to maintain satisfactory firebox conditions.
- Crude-oil side convection outlet temperatures were lower.

Here is what happened. Preheating the air by 300°F had raised the burner flame temperature by 300°F. The hotter flames then radiated more heat per pound of fuel consumed, and as a result, the firebox became much hotter. The same factor that reduces fuel consumption in the radiant section also reduced the flow of flue gas to

the convective section, thus reducing the convective-section outlet temperature.

So what can we do to actually run more crude, especially as there is no bypass around this new air preheater?

The answer, of course, is to allow more air into the firebox and thus generate more flue gas. The firebox oxygen was increased from 3 to 6 percent, which reduced the firebox temperature to its prerevamp state. This, in turn, increased the pounds of flue gas flowing through the convective section and increased the heat absorbed in the convective section.

The increased combustion air flow was needed not for combustion, but to transfer heat from the radiant section (firebox) to the convective section. This is what we call *heat balancing*. In this situation, oxygen requirements to reach absolute combustion become irrelevant as we are now operating with a very plentiful supply of oxygen.

29.7.5 Lack of Air Affects Draft and Combustion

Normally, opening the air registers at the base of a heater reduces draft (i.e., increases pressure) below the bottom row of convective tubes. This happens because the extra combustion air creates more flue gas, which results in an increased ΔP across the convective-section tubes and the stack damper.

However, if the heater is operating in an oxygen-deficient environment, or it is running below the point of absolute combustion, draft will likely increase as the air registers are opened. We saw this happen recently. When the air supply to the burners was increased, the flame appearance changed from dull orange to a bright bluish-yellow. The firebox became noticeably hotter. We assumed (but failed to check) that the stack also became hotter. A hotter stack reduces the density of the flue gas, and that will always increase draft. The air flow was normally restricted to meet an arbitrary—and apparently excessively low—O₂ target in the stack.

The conclusion, then, is that if opening air registers increased draft, the heater had been operating below the point of absolute combustion.

Reference

1. T. E. Butterfield et al., *Steam and Gas Engineering*, 4th edition (New York: Van Nostrand, 1947), pp. 205–206.



A couple of fellow combatants: Norm Lieberman (coauthor) and Tom Germany (plant manager, Calumet Refinery, Shreveport, Louisiana), 30 years after solving the combustion air limit on a fired heater at the Tenneco Refinery in New Orleans. The cause of the limit they found was an air-inlet screen fouled with plastic sheeting; the equipment they used is seen in Fig. 29.7. Norm is now older, Tom is wiser, but neither has lost sight of the need to work problems through in the field, close to the equipment.

CHAPTER 30

Fired Heaters: Process Side

Coking Furnace Tubes and Tube Failures

The heat of combustion in a process heater may liberate 100×10^6 Btu/h. If the heater's efficiency is 78 percent, the heat absorbed by the tubes should be 78×10^6 Btu/h. When observing a heater's operation, it is a good idea to check this heat balance by calculation.

30.1 Process Duty versus Heat Liberation

The heat of combustion is a product of the amount of fuel consumed and the net heating value of the fuel. The heater's efficiency is a function of the flue-gas stack temperature, the excess air or oxygen, and the ambient-heat losses from the firebox and the convective-section structures.

The heat absorbed by the tubes is the sum of the heat of vaporization of the process liquid to vapor, plus the increase in the sensible-heat content of the flowing process fluid.

If the heat picked up from the combustion of the fuel does not equal the heat absorbed by the process fluid, then something is amiss with the data. Often, determining the cause of such an inconsistency will reveal several fundamental operating or measurement problems with a fired heater. Quite commonly, we may find that the metered fuel-gas rate is wrong, or that the ambient-heat losses are much greater than anticipated in our calculations.

30.1.1 Distribution of Heat of Combustion

When fuel is burned in a fired heater, there are three major products of combustion:

- Water
- Carbon dioxide
- Heat or British thermal units (BTUs)

The water plus the CO_2 mix with the inert nitrogen in the combustion air to form flue gas. There is also some oxygen in this flue gas. This oxygen is called *excess* O_2 . A typical excess O_2 content of flue gas is 2 to 6 percent. About 80 percent of the flue gas is N_2 . The rest is H_2O plus CO_2 .

The heat liberated by combustion is distributed in one of three ways:

- Some of the heat is used to increase the temperature of the combustion air from the ambient-air temperature to the temperature of the flue gas leaving the heater's firebox. This heat is called the *convective-heat* content of the flue gas.
- Some of the heat is radiated directly to the heater tubes.
- The rest of the heat is radiated to the refractory walls.

In most heaters, the majority of the heat of combustion is radiated to the refractory walls. The glowing refractory walls then reradiate the heat to the heater tubes.

Many older heaters have massive brick refractory walls. The weight of these walls greatly exceeds the weight of the heater's tubes. The massive brick refractory walls store a great deal of heat. This creates several operating problems. One such problem is that it takes many hours to bring such a heater up to its normal operating temperature. However, a far more serious problem occurs when the process flow to a heater is interrupted.

Newer heaters typically have thin reflective tiles, rather than massive refractory brick walls. Such newer heaters will heat up more rapidly. Also, the process fluid outlet temperature responds more rapidly to changes in the firing rate. This improves the heater outlet temperature control. Perhaps for this reason, it seems that heaters with reflective refractory walls are less subject to process tube coking and shortened heater run lengths. There is also a process, called "alonizing," that increases the reflectivity of older brick refractory heater walls.

30.1.2 Reradiation of Heat from Refractory

When flow is first lost, the fuel to the heater is often automatically tripped off. But the refractory walls continue to radiate heat to the process tubes. It is rather like a large flywheel. Even after we stop

cranking the wheel, the rotational energy stored by the wheel keeps it spinning. Even when all the firing is stopped, the energy stored in the refractory walls continues to radiate heat to the process tubes.

But without flow, there is no way to carry the heat away from the tubes. Therefore, the tubes overheat. The temperature of the tubes may approach the temperature of the refractory, at the point in time when flow is lost. The refractory temperature is indicated by the firebox temperature or the temperature of the flue gas flowing from the firebox into the convective section.

A typical firebox temperature is 1500°F. Thus, the heater tubes can reach 1300°F on loss of the process flow, even though the fuel flow has been immediately stopped. Tubes with a low chrome content may bend and distort as a result of such overheating. Even at 1000°F, residual liquid left in the tubes when flow is lost may thermally degrade to a carbonaceous solid or heavy polymer that fouls the interior of the tubes.

One way to combat this problem is with steam. As soon as the flow is interrupted, high-pressure purge steam is automatically opened into the heater tube inlets. The steam blows the residual liquid out of the tubes and also helps remove heat from the tubes.

30.1.2.1 Stuttering-Feed Interruption

After the purge steam begins to flow, the refractory walls and the tubes slowly cool. However, if process flow is reintroduced to the heater during this cooling period, a serious problem may result. The first few gallons of process liquid flowing through the heater will become extremely hot. The liquid may get so hot that it will turn to solid coke and partially plug the heater tubes.

If this problem—the sudden loss of flow, followed by the premature restoration of flow—occurs repeatedly over a period of a few hours, then layers of fouling deposits or coke are accumulated inside the tubes until a heater shutdown becomes unavoidable. This sort of failure is called a *stuttering-feed interruption*.

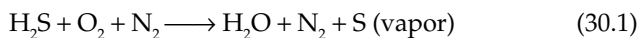
Modern technology has come a long way in mitigating such problems. In new heaters, lightweight ceramic tiles, rather than massive brick refractory walls, are the norm. These ceramic tiles do not store very much heat. Hence, when the heater process flow is reduced or lost, as long as the fuel flow is quickly curtailed, the tubes tend not to overheat.

30.1.3 Adiabatic Combustion

In 1980, there was a strike at the Amoco Oil Refinery in Texas City. I was assigned during this emergency to work as the chief operator on the sulfur plant. You see, I was the engineer who had designed the sulfur plant. Therefore, Larry Durland, the refinery manager, thought that I would be the logical person for the chief operator's job.

To be honest, I was a scab. I know I was a scab, because when the hourly operators returned to work, they gave me a gift: a T-shirt. On the front it said "SCAB." On the back it said, "I like cheese," meaning I was a rat!

A sulfur plant converts H_2S to elemental sulfur through the partial oxidation of H_2S :



This reaction takes place in an *adiabatic-combustion chamber*, shown in Fig. 30.1. This chamber has no tubes to absorb radiant heat. Plenty of radiant heat is liberated, but only to the refractory brick walls. The bricks then reradiate the heat back into the gaseous products of reaction. This is called *adiabatic combustion* because no heat is lost from the combustion reaction to radiation. The adiabatic-combustion temperature for the preceding reaction [Eq. (30.1)] is about 2300°F. The refractory used to contain this high temperature is manufactured from 90 percent alumina. Such refractory may be exposed to temperatures of up to 2900°F without damage.

During the strike, the sulfur plant was shut down for minor repairs. I had to supervise its start-up. Mainly, I had to reheat the adiabatic-combustion chamber to 1800°F before restoring the flow of H_2S . This was done by burning a controlled amount of methane or natural gas with a carefully regulated flow of air. The idea was to slowly heat up the combustion chamber with hot flue gas by 100 to 200°F per hour. This slow reheat was needed to avoid cracking the refractory bricks because of uneven heating. To carry away a portion of the heat of combustion of the natural gas, we used pipeline nitrogen.

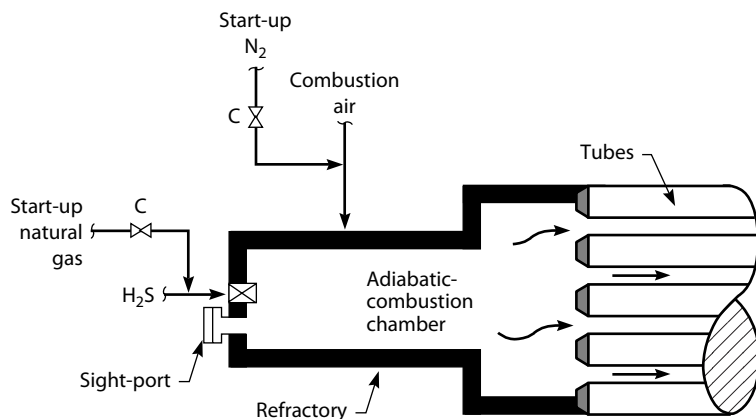


FIGURE 30.1 Front end of a sulfur recovery plant.

I thought it best to control the flows of nitrogen, natural gas, and air, myself. Basically, the other scabs at the sulfur plant were head office people, whom I did not trust. The reheat phase of the start-up seemed to be going quite slowly. The combustion chamber temperature crept up by 50°F an hour, rather than the normal 150°F per hour. To speed the reheat, I reduced the nitrogen flow. This helped, but not by much.

It all seemed so odd. Especially as the interior of the chamber, viewed through the sight-port (see Fig. 30.1), had a dazzling white appearance. I remember thinking, "I cannot spend the rest of my life out here. I will have to speed this along." So I shut off the nitrogen and turned up the gas and air. Finally, the chamber temperature started to climb at a respectable rate. But then I began to see something in the chamber that was utterly impossible.

Of course, peering through the glass sight-port, it is hard to see very well. But it seemed to me as if the opposite refractory wall was very slowly, but still perceptibly melting. But this could not be, for several good reasons:

- *Reason number one:* The 90 percent alumina refractory was rated for 3000°F service.
- *Reason number two:* The indicated combustion chamber temperature was still only 1500°F.
- *Reason number three:* I am a nice person, and bad things should not happen to nice people.

After two hours, it became clear that the openings in the opposite refractory wall were shrinking. These openings permitted the hot flue gas to exit the combustion chamber and flow into the tubes of a heat-recovery boiler. The apparently melting refractory was sagging and restricting these apertures.

I noticed, with a sinking heart, that the combustion chamber pressure was steadily rising. It rose from 2 to 12 psig, at which point the fuel gas tripped off as a result of the high pressure. The sulfur plant start-up was aborted! But what had happened?

An investigation showed that the *thermowell*—a steel or ceramic tube containing the thermocouple wires—was not fully inserted into the adiabatic-combustion chamber. The end of the thermowell was only half way into the 12-inch-thick refractory wall, as shown in Fig. 30.2. Therefore, the thermocouple was measuring the relatively cool zone inside the refractory wall, rather than the far hotter zone in the combustion chamber. I should have paid closer attention to the physical appearance of the chamber. Bricks glowing bright red are radiating heat at about 1500°F. Bricks glowing a dazzling white are radiating heat above 2800°F (see Table 30.1).

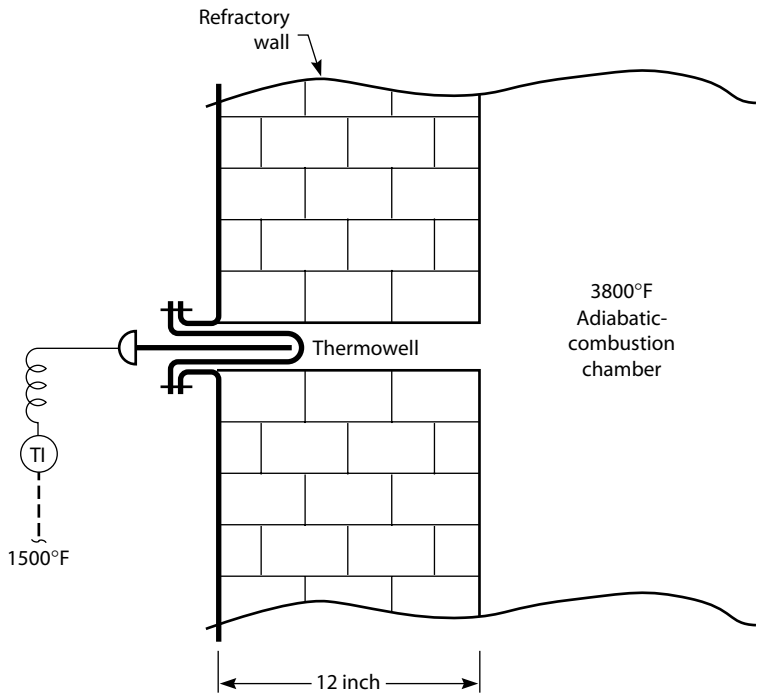


FIGURE 30.2 Overheating a firebox because of shortened thermowell.

Color	Temperature, °F
Black red	990
Dark red	1050
Dark cherry	1250
Bright cherry	1375
Light red	1550
Orange	1650
Light orange	1725
Yellow	1825
Light yellow	1975
White	2200
Dazzling white	2730

TABLE 30.1 Visual Estimation of Temperatures

How hot did the refractory surface get? This value can be calculated if we assume the following:

- Fuel was 100 percent methane.
- Complete combustion (implying theoretically perfect air/fuel gas mixing).
- No excess oxygen.
- No heat losses to surroundings.
- Air is available at 60°F.
- Pipeline nitrogen shutoff.

Then the calculated temperature of the combustion chamber is 3800°F. This is called the *adiabatic flame temperature*. Such a temperature is quite sufficient to turn even bricks into a high-viscosity, lava-type, semisolid fluid.

Larry Durland, the refinery manager, was not particularly pleased. He unjustly blamed me for this incident. Just because I had designed the sulfur plant, was the chief operator, and had been personally restreaming the sulfur plant, he said it was all my fault, and that I should have known better.

30.2 Heater Tube Failures

Heater tubes are designed to operate at a particular pressure and temperature. The design pressure of the tube is not the inlet operating pressure of the heater. The design tube pressure is the heater charge pump dead-head, or shut-in, pressure, as discussed in Chap. 34. The design temperature of the tube is not the heater outlet process operating temperature. The design tube temperature is the anticipated or calculated maximum *tube skin temperature* (at end-of-run conditions), which is simply the temperature of the exterior metal surface of the tube. Many plants call this temperature the *tube metal indication* (TMI).

The calculated tube skin temperature is mainly a function of the fouling resistance assumed inside the tube. The greater the assumed fouling resistance, the higher the design tube skin temperature, and the thicker the tube wall. In a sense, then, we partially assume the design tube thickness, on the basis of experience, for a particular plant service.

A typical process heater tube diameter is 4 to 10 inch. Tube thickness is usually between 0.25 and 0.50 inch. Heater tubes are often constructed out of chrome steel. A high chrome content is 13 percent. The chrome content increases the heat resistance of the tube. A tube with a 11 to 13 percent chrome content can normally withstand a skin temperature of up to 1300 to 1350°F. A low-chrome-content tube of perhaps 3 percent may be limited to 1200°F tube metal temperature.

Naturally, the pressure, thickness, and diameter of the tube all affect its maximum skin temperature limitations.

For added corrosion and temperature resistance, the nickel content of tubes and sometimes the Moly (molybdenum) content as well are increased. Tubes with a high nickel content are classified as 300 series stainless steels. A 0.5 percent silicon content is used to enhance the tube's oxidation or exterior scaling resistance.

I had a job working in India last year. The unit bottleneck that I was consulting for was the furnace. Both the firing rate and TMI temperature were low. However, the tubes themselves were ordinary carbon steel (c.s.). Not only the convective tubes, but the radiant tubes, all were carbon steel. Theoretically, engineering calculations showed c.s. was adequate. Note that the furnace specs were issued by an American contractor, and not the locals.

Theory is fine. But it needs to be tempered with common sense. Process heaters in refinery hydrocarbon service should employ chrome steel and not c.s. tubes in the radiant wall and shock tubes. Carbon steel tubes should be confined to the convective section.

30.2.1 High-Temperature Creep

When the tube metal temperature exceeds a value of 1300 to 1400°F, it becomes plastic. This means that the pressure inside the tube causes the tube diameter to expand. This is called *high-temperature creep*. As the diameter of the tube bulges and expands, the tube walls become progressively thinner and ultimately too thin to constrain the pressure inside the tube, and the tube bursts. Large-diameter tubes operating at higher pressures and with a thin wall thickness fail at a relatively low tube skin temperature.

Tubes seldom fail because of external oxidation, and tubes rarely "burn up." They fail because of high-temperature creep, which causes the tube to expand and burst. Thus, the fundamental cause of tube failure is a high localized temperature, which is called a "hot spot."

30.2.2 Purge Steam

When a heater tube fails, the process fluid spills out into the firebox. Let's assume that the process fluid is a combustible liquid. Will the leaking process fluid burn? There is probably not enough excess oxygen in the firebox to support a substantial amount of additional combustion.

Flames and black smoke will pour out of the heater's stack. It looks very dangerous, but it is really not. There is not enough excess oxygen in the firebox to cause a very high temperature. The combustion gases, or flue gas, are too fuel-rich to explode. We say that the flue gas is above the *explosive region*.

To prevent the flue gas from exploding, we need to proceed cautiously. If we just block-in the leaking process tubes and fuel-gas supply,

the fuel gas content of the flue gas will gradually decrease. The air:fuel ratio in the flue gas will increase until the flue-gas composition enters the explosive region. If the firebox refractory walls are still hot enough to initiate combustion, the firebox will now explode.

This happened on a unit I was supervising. A score of operators were lying on the concrete with their polyester shirts burning. This incident happened in 1982, before the need to wear fire-retardant Nomex coveralls was recognized.

The correct way to prevent this sort of firebox explosion is to use firebox purge steam. A typical heater firebox might have a half dozen purge steam connections. The idea is to displace the air in the firebox with steam. Then the flows of combustible process liquid and fuel gas may be safely stopped without fear of entering the explosive region. This purge steam is different from the coil purge steam used in the heater's tubes.

Purge steam is sometimes called *snuffing* or *box steam*. I have a few rules about the design and operation of this system to ensure its safe use:

- Locate the steam valves at least 30 meters back from the heater.
- The steam line must be heat traced and insulated to prevent water accumulation.
- Steam traps both up and downstream of steam snuffing valves are needed.
- These valves must be greased and exercised once or twice a year. Valves should have grease nipples.

Blowing steam condensate into an 1800°F firebox with a failed tube blowing out vacuum resid is best avoided. Your ability to train the operators as to how to react in these sorts of emergencies is the test of a good supervisor.

30.2.3 Identifying Thin Tubes and Hot Spots

A bulging tube indicates a thin area of a tube. If the diameter of a tube were to uniformly increase by 20 percent, then the thickness of the tube would decrease by 20 percent. But tubes rarely expand uniformly. They expand mainly on that side of the tube that is hottest. Hence, a tube bulge that increases the overall tube circumference by 20 percent typically reduces the thickness of the tube in the area of the bulge by 40 percent. For many tubes, this reduction in thickness is sufficient to cause tube failure. There is no theoretical basis for these statements. It is just a summary of what I have seen when a section of tubing is cut from a heater for failure analysis (see Fig. 30.3).

The best way to find thin, bulging tubes when the heater is offline is to run a metal ring down the length of a tube. A severe

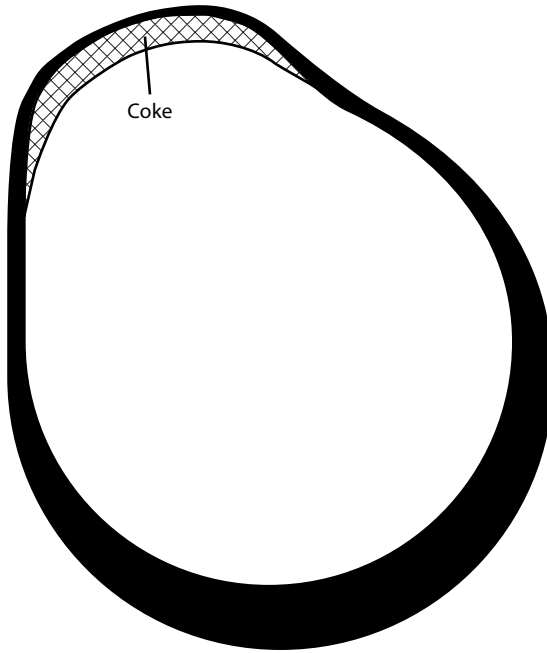


FIGURE 30.3 Heater tube cross-section close to failure.

bulge may be visually observed even when the heater is in service. Sagging tubes do not represent thin tubes. I have seen radiant-section, horizontal tubes sag by more than one tube diameter and operate for years without failing.

30.2.3.1 Identification of Hot Spots

Localized overheating of a tube causes localized high-temperature creep. This leads to the plastic deformation of a tube and hence thin tube walls. Such hot spots are indicated by the color of the tube, as shown in Table 30.1. This chart is not a function of the material of the tube. Carbon steel tubes, as well as tubes of all alloys, glow with a color that corresponds only to the localized tube temperature. This is the principle of operation of the *optical pyrometer*, a device that is widely used in the process industry to remotely measure tube skin temperatures.

Sometimes, deposits accumulate on the exterior of tubes. These deposits are unavoidable when heavy industrial fuel oil is fired. Such fuel oils contain high concentrations of vanadium, nickel, sodium, and iron (in that order of concentration). These metals deposit as an ash on the exterior of both the radiant- and convective-section tubes. The ash will get very hot and will turn sections of tubes yellow or even whitish silver. This is not an indication of tube overheating. If the

deposits get thick enough, they will interfere with the rate of heat absorption by the tube. It is very difficult to distinguish between a real hot spot and glowing ash. Dirty fuel gas also leads to ash formation on the radiant surface of tubes.

Fuel oils that contain both vanadium and sodium are a special problem. I've discussed this problem in a later example dealing with corrosion. The difficulty is that a mixture of 99 percent vanadium and 1 percent sodium forms a eutectic mixture, which melts at about 1300°F. The viscous liquid that results is corrosive. I've seen rapid rates of localized tube thinning on vertical, radiant section of tubes.

If the deposits don't melt, they will accumulate on the surface of the radiant section tubes. I've seen an 0.125 inch of vanadium evenly coating a 300 mm Btu/hr heater. A heater flux rate of 14,000 Btu/hr/ft² had been sustained for five years with no tube damage. The vanadium acted as a protective coating for the tubes, except where it melted and caused a tube leak.

30.2.3.2 What Causes Hot Spots?

Four conditions will promote the formation of hot spots on tubes:

1. *Flame impingement.* This is often caused by dirty burner tips, lack of combustion air, poorly designed burners, high burner tip pressure, improper adjustment of the burner, or improper draft. I have seen a heater in Cartagena, Colombia, with the flames being forced outward against the upper radiant wall tubes. The problem was an extreme positive pressure in the firebox, due to excessive pressure drop of the flue gas in the fouled convective section.
2. *Poor radiant-heat distribution.* A high localized *radiant-heat flux* (in Btu/[h(ft²)] of tube surface area) will result in a hot spot. A moderate average radiant-heat flux for a process heater might be 12,000 Btu/[h(ft²)]. However, if heat distribution is poor, the local radiant-heat flux could be 20,000 Btu/[h(ft²)]. This is excessive for most services. The cause of the poor radiant-heat distribution could be sagging tubes, too small a firebox for the heat released, or inadequate tube spacing. Having many small burners rather than just a few large burners promotes more even radiant-heat distribution. Figure 30.4 shows a double-fired heater. This type of heater results in excellent heat distribution and can tolerate average radiant-heat fluxes 15 to 25 percent higher than conventional heaters with radiant wall tubes.
3. *Interior tube deposits.* Coke, salts, and corrosion products can adhere to the inside wall of the tube. The deposits precipitate out of the flowing fluid, due to low tube-side velocity. Once the solids stop moving, they bake onto the tube wall. The flowing fluid can no longer effectively cool that portion of the tube wall

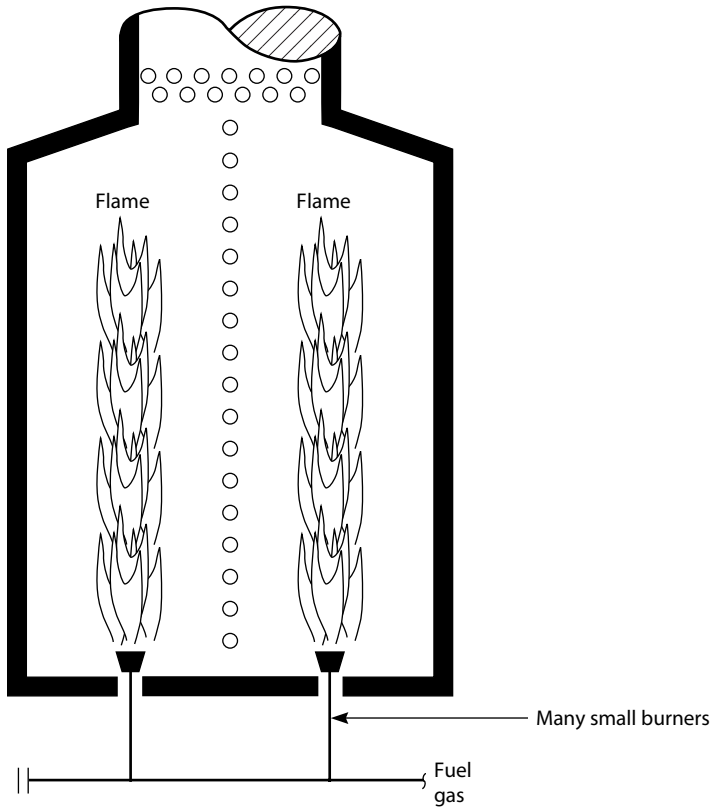


FIGURE 30.4 A double-fired heater.

covered with these internal deposits. The minimum acceptable tube-side velocity to retard solids sticking to the tube wall really depends on the service. A few general guidelines are

- Mixed-phase vapor and liquid flow, 20 ft/s
- Liquid-only flow, 8 ft/s

When a tube that has failed because of a hot spot caused by a coke deposit inside is cut in half, we would observe a pattern similar to that shown in Fig. 30.3.

4. *Dry-point deposits.* We sometimes see that a certain heater tube will glow a light red and fail for no apparent reason. The particular tube position in the firebox seems to be far more subject to failure than its neighbors. The tube is not located in an area of flame impingement, and the tubes upstream and downstream are a nice dark red (see Table 30.1).

This strange behavior is likely due to the heater feed going through its dry point in this tube position. Let's say that the heater feed is 10 weight percent (wt.%) liquid. The heater effluent is 100 percent vapor. Unfortunately, the liquid portion of the heater feed is likely to contain small amounts of dissolved, or suspended, solids. The concentration of these solids increases in the liquid as the liquid vaporizes. When the liquid dries out, the solids stick to the wall of the tube. The radiant heat absorbed by the tube can no longer be efficiently removed from the tube wall by the flowing vapor. The tube overheats and bulges, the walls thin at the bulges, and the tube bursts.

30.3 Flow in Heater Tubes

30.3.1 Loss of Flow

Let us say we have an ordinary orifice-type flowmeter, as shown in Fig. 30.5 (see Chap. 10, "How Instruments Work"). What happens if the low pressure (i.e., the downstream) orifice tap plugs? Does the indicated flow go up or down?

If a pressure tap plugs, the measured pressure will decrease. The measured pressure difference across the orifice plate will increase.

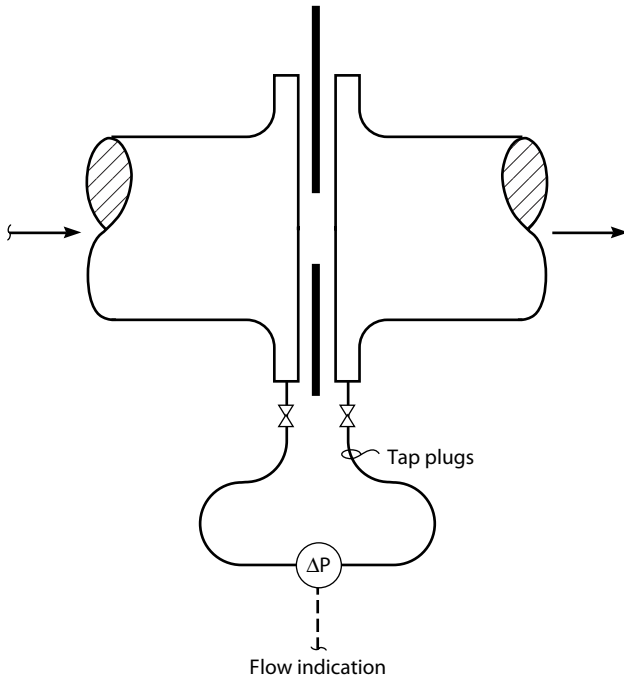


FIGURE 30.5 Plugged low-pressure tap increases indicated flow.

The indicated flow will go up. If the flowmeter is controlling a flow-control valve on a tube inlet to a heater, the valve will then close. Flow to the tube inlet will be lost. The tubes downstream of this flow-control valve will likely be damaged because of overheating, or they will plug and foul as a result of thermal degradation of the process fluid.

One way to limit the damage due to this all-too-common problem is to limit the amount the flow-control valve can close on automatic control to, say, 20 percent open. Make sure, though, that with the valve 20 percent open, enough flow is sustained through the tubes to prevent tube damage due to too low a process flow.

30.3.2 Annular Flow

Sometimes the liquid flow through a heater tube is low. Low flow promotes thermal degradation of the liquid and harmful buildup of deposits inside the tubes. To combat this problem, we often add *velocity steam* to the heater's tubes to increase the linear velocity of the oil. However, if the physical properties, especially the density, of the oil and steam are very different, phase separation will result. This means that the low-density steam will run down the center of the tube. The higher-density liquid will creep along the periphery of the tube at a much lower velocity. This sort of phase separation is called *annular flow*. When a tube is in this flow regime, increasing steam velocity further does very little to increase the liquid velocity or in combating the thermal degradation of the liquid.

30.4 Low-NOx Burners

NOx stands for a variety of nitric oxides. Many heaters in the United States have been retrofitted with staged burners in the last decade or so. These staged burners combust the fuel in two or three stages. For example, in the burner shown in Fig. 30.6, 50 percent of the fuel is burned with 100 percent of the air. The flame produced by this first stage of combustion radiates heat to the process tubes and refractory walls. Next, the remaining 50 percent of the fuel is added around the circumference of the first-stage burner. This second stage of combustion also liberates radiant heat. But because the radiant heat is liberated in two steps, the maximum flame temperature is reduced. This has two favorable results:

1. Nitric oxide (NOx) production is reduced, as the oxidation of nitrogen is a strong function of the flame temperature. Hence the term *low-NOx burners*.
2. Heat liberation from the low-NOx burner is more uniform than with a conventional burner. This permits a higher average firebox temperature to be sustained, without promoting

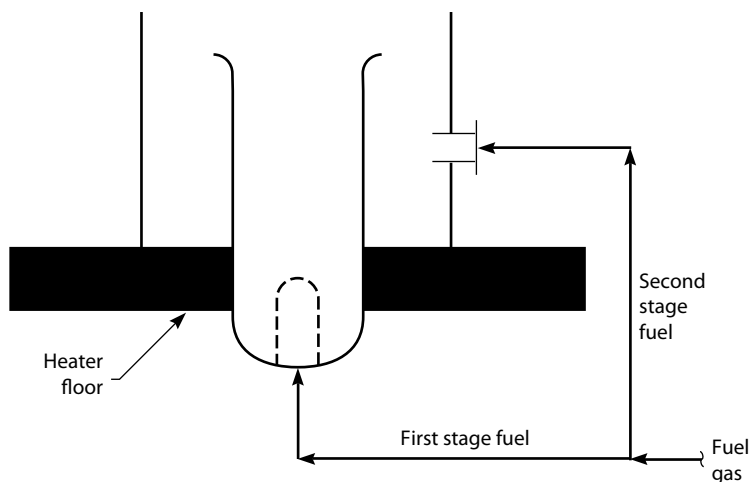


FIGURE 30.6 A two-stage low NO_x burner.

hot spots on the tubes and interior tube coking. I have recently seen a refinery crude unit vacuum tower feed heater expanded by 15 percent, simply by retrofitting with two-stage, low-NO_x, fuel-gas burners. These burners replaced old-fashioned burners that did a poor job of mixing the air and fuel gas.

Premix, or primary-air, burners do a great job of mixing air and fuel gas. But they also produce a high flame temperature and hence, higher concentrations of NO_x in the heater's effluent flue gas.

30.5 Tube Fire-Side Heaters

So far, we have been discussing fired heaters, with the fire outside the tubes; that is, the fire is outside the heat-exchange surface. Many fired heaters have the fire, or at least the hot flue gas, inside the heat-exchange surface.

One common example of firing on the tube side of a heater is the glycol-regeneration boiler, shown in Fig. 30.7. This type of heater will typically have a high excess O₂ to prevent high flame temperatures, which could overheat the fire tube. Also, the fire tube is kept submerged in liquid to prevent tube overheating.

In multiple-tube boilers (usually horizontal) the fire may be on the tube side. As long as the fire tubes are kept submerged in water, the tubes do not overheat. Boilers of this type are widely used in the regeneration of spent sulfuric acid, and in the production of elemental sulfur from hydrogen sulfide.

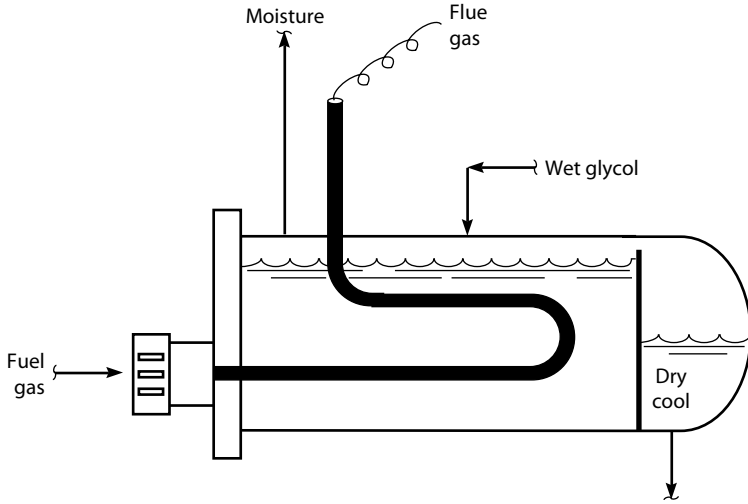


FIGURE 30.7 Glycol regenerator.

The great difficulty with such fire-side, multitube boilers is overheating the hot-side tubesheet. This tubesheet is exposed to the full temperature of the combustion gases, but is difficult to keep cool with the boiler's feedwater. Also, the ends of the tubes, where they are sealed in the tubesheet, are hard to keep cool. To protect these ends from the direct radiant heat in the combustion chamber, ferrule inserts, about 2 inches long, are cemented into the front end of each tube.

If these ferrules fail, or if the tubesheet overheats, the ends of the tubes will pull away from the tubesheet. The result is called a "roll leak." Boiler feedwater will blow out of these roll leaks. Rerolling and seal welding the ends of the tubes, while difficult, is then the only way to stop such leaks.

Old style steam locomotives were boilers on wheels with the fire on the tube side. I visited one just last month. Abandoned on a railroad siding in New Orleans 50 years ago, it almost looked ready to raise steam again. Its tubesheet was visible from the fireman's door.

CHAPTER 31

Refrigeration Systems

An Introduction to Centrifugal Compressors

Waste heat must occasionally be removed from a process at below-ambient temperatures. This is normally accomplished by evaporating a light fluid called a *refrigerant*. Common industrial refrigerants are

- Ammonia
- Propane
- Freon

Your home central air-conditioning system is a good example of a simple refrigeration loop. Figure 31.1 illustrates the basic components:

- Refrigerant motor-driven compressor
- Refrigerant condenser
- Refrigerant letdown valve
- Evaporator
- Refrigerant receiver

The basic process flow of the refrigerant system is:

1. The compressed refrigerant vapor is discharged from the compressor. This vapor is *superheated*, meaning that it is above its dew-point temperature.
2. The vapor is next cooled and condensed in the refrigerant condenser. Typically, 75 to 80 percent of the condenser heat duty is the latent heat of condensation of the refrigerant. The remaining heat duty is sensible-heat removal. The sensible heat

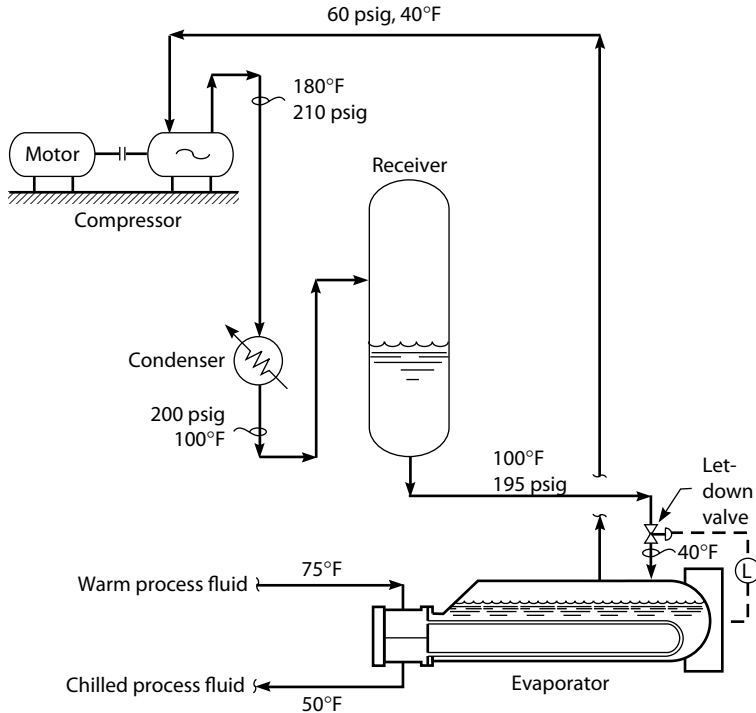


FIGURE 31.1 Refrigeration system.

removed is a combination of desuperheating the vapor and subcooling the condensed liquid refrigerant.

3. The subcooled refrigerant next flows into the refrigerant receiver. *Subcooled* means that the refrigerant temperature is below its boiling point or bubble-point temperature. The liquid refrigerant loses a small amount of pressure as it enters the receiver. The pressure loss is due to friction in the piping and an increase in the elevation of the refrigerant as it enters the receiver.
4. The condensed liquid refrigerant next flows through the *letdown valve*. In Fig. 31.1, this letdown valve is not controlling the receiver liquid level. The receiver level is allowed to vary as the total refrigerant inventory changes. This idea can be better understood if we remember that this is a closed system. That is, we have to take swings in the refrigerant inventory somewhere in the system.
5. The refrigerant liquid partially flashes to a vapor as it flows through the letdown valve. The flashing represents the conversion of the sensible heat of the refrigerant to latent heat of vaporization. In Fig. 31.1, the refrigerant is chilled from

100 to 40°F. Approximately 25 percent of the liquid flashes to a vapor to provide this autorefrigeration.

The compressor, motor, condenser, receiver, and letdown valve are all components of your home central air-conditioning unit. They are installed as a package, surrounded by the condenser, outside your house. The evaporator is located in your attic. To continue our description of the process flow:

6. The partially vaporized refrigerant flows into the evaporator. In Fig. 31.1, the evaporator shown is similar to a kettle-type reboiler (see Chap. 8). The process fluid flows through the tube side of the kettle evaporator. The refrigerant liquid level is maintained by the letdown valve. The refrigerant vapor flows from the top of the kettle to the compressor suction.
7. The motor-driven compressor boosts the pressure of the refrigerant vapors from the evaporator pressure up to the condenser pressure.

31.1 Refrigerant Receiver

Looking at Fig. 31.1, the need for the refrigerant receiver is not immediately obvious. It does not appear to serve any process function. Many refrigerant systems are constructed without this vessel. However, it does have several important uses. For one thing, it provides a quiet zone for entrained lubricating and seal oils to settle out.

Lubricating and seal oil often leak into the circulating refrigerant. The lower the suction pressure of the compressor, the greater the problem. Eventually, the amount of heavy oil that accumulates in the evaporator will cause a problem. The heavy oil is largely insoluble in the refrigerant. As the refrigerant boils away out of the evaporator, the heavy oil is left behind. This oil partly coats the refrigerant tubes. The heat-transfer efficiency of the evaporator is reduced.

Turbulence caused by the boiling of the refrigerant prevents efficient draining of heavy oil from the evaporator. However, if the refrigerant receiver has been designed properly, it can be used to trap out the heavy lubricating oil.

31.1.1 Inventory Control

In Fig. 31.1, there are two levels that must be controlled: the evaporator level and the refrigerant receiver level. It is impossible to control both levels with two level-control valves. One level must be held with a level-control valve. The other level must be held by adjusting the system liquid refrigerant inventory. It is usually better to add makeup refrigerant to keep a decent liquid level in the receiver and control the flow of refrigerant into the evaporator. However, reversing this control scheme by adjusting inventory control in the evaporator is also possible.

31.1.2 Vapor Trap

Another way of looking at the receiver vessel is to think of it as a giant steam trap. As long as the liquid level is maintained in the receiver, uncondensed refrigerant vapors cannot escape from the condenser. Let's assume that such vapors did escape from the condenser. If there were no refrigerant receiver vessel, these vapors would blow through the let-down valve. Reducing the pressure of the vapors produces just a tiny bit of refrigeration as compared to vaporization of liquefied refrigerant.

Now let's assume that there is a refrigerant receiver between the condenser and letdown valve. This traps any uncondensed vapors. The accumulation of these vapors raises the pressure in the receiver. This puts back-pressure on the condenser. The higher condenser pressure promotes more complete condensation of the refrigerant vapors.

Finally, the receiver will accumulate any non-condensable (or hard-to-condense) components that have accidentally entered the system. Air left in the vessels on start-up is one such example. Traces of methane and ethane in a propane refrigerant system are another. These light vapors may be vented from the top of the receiver during normal operations.

31.2 Evaporator Temperature Control

The system shown in Fig. 31.1 has no provision for controlling the evaporator temperature; this is controlled solely by the compressor suction pressure. The lower the compressor suction pressure, the lower the evaporator temperature. This is exactly how our home air conditioner works. If the evaporator temperature is too cold, what can we do? Well, if this were a steam turbine compressor, gas engine drive, or any other type of variable-speed driver, we could reduce the compressor's speed. This would reduce the flow of refrigerant, and raise both the evaporator and compressor suction pressure.

But we are working with an ordinary AC (alternating-current) motor—which is a fixed-speed device. There are then three methods available to control the temperature in the evaporator: spillback, discharge throttling, and suction throttling.

31.2.1 Spillback

Spillback bypasses the compressor discharge gas back to the compressor suction. This is a relatively energy-inefficient way to increase the evaporator temperature. Spillback is discussed in Chap. 41, "Centrifugal Compressors and Surge."

31.2.2 Discharge Throttling

Discharge throttling is also an energy-inefficient way to control the evaporator temperature. Even worse, if we are working with a centrifugal compressor, discharge throttling may cause the compressor to *surge*.

31.2.3 Suction Throttling

Suction throttling is the preferred method, as shown in Fig. 31.1.

Suction throttling also wastes energy. Whenever a control valve is partially closed in any service, energy is always wasted. But with a fixed-speed centrifugal compressor, it is still the best option. As the suction valve is closed, two things happen. The flow of refrigerant vapor to the compressor is reduced, and the compressor suction pressure drops. The reduction in the refrigerant vapor flow reduces the workload on the compressor. But the reduction in the suction pressure increases the *compression ratio* (discharge pressure divided by suction pressure). This increases the workload on the compressor.

As we will discuss in Chap. 41, increasing the compression ratio is a small effect on compressor workload as compared to decreasing the vapor flow as long as we are operating on the flat part of the compressor's flow vs. the head performance curve, which is almost always the case for a centrifugal machine. Hence, suction throttling will significantly reduce the horsepower load on the compressor, as well as increase the evaporator temperature for a motor-driven compressor.

Suction throttling is usually done with a "butterfly" control valve, which has a very low pressure drop when it is wide open.

31.3 Compressor and Condenser Operation

31.3.1 Compressor Operation

Let's say we have a centrifugal refrigeration compressor driven by a motor. The motor is tripping off because of high amperage. Should we open or close the suction throttle valve shown in Fig. 31.2? Answer—close it. Of course, both the evaporator vapor outlet temperature and the process fluid outlet temperature will increase. But that is the price we pay for having too small a motor driver on the compressor. Does this mean that when our home air conditioner gets low on freon, our electric bill drops? Correct. But the price we pay is a hot home.

How about the compressor discharge pressure? What is this controlled by? Answer—the condenser. The compressor discharge pressure has nothing whatsoever to do with the compressor. And this statement is true regardless of whether the compressor is a reciprocating or centrifugal machine, or fixed- or variable-speed. It is the condenser temperature that determines the compressor discharge pressure.

Lowering the compressor discharge pressure can be achieved by providing cooler water to the refrigerant condenser. But how does this affect the horsepower load or electrical power demand on the compressor's

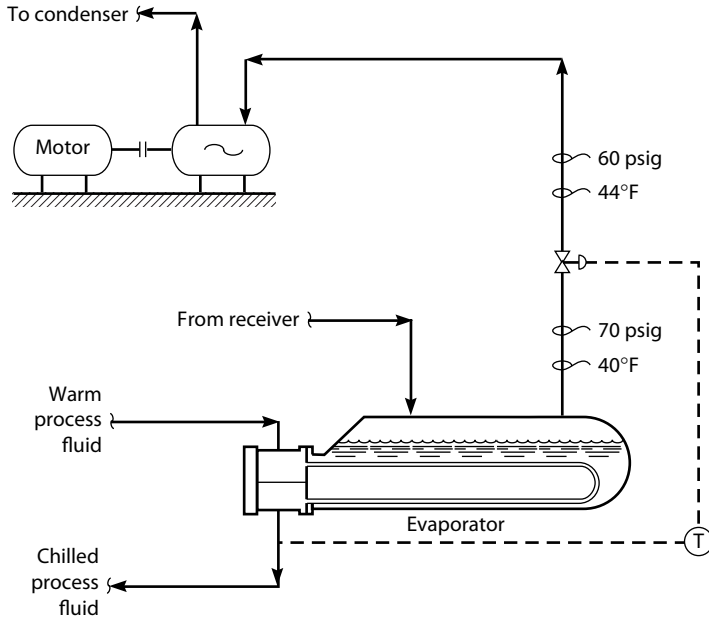


FIGURE 31.2 Suction throttling to control evaporator temperature.

driver? It depends on how much the refrigerant vapor flow increases, as the compressor discharge pressure is reduced.

For instance, suppose we are working with a reciprocating compressor. This particular machine has a very small volumetric clearance and a very high volumetric efficiency (these terms are explained in Chap. 42, “Reciprocating Compressors”). With this compressor, a significant decrease in discharge pressure will result in only a tiny increase in vapor flow. Therefore, the work or amp load on the driver will decrease.

On the other hand, suppose we are working with a centrifugal compressor. This particular machine has a flat operating curve. This means that a small decrease in discharge pressure will result in a large increase in vapor flow. Therefore, the load on this compressor’s driver will increase.

This discussion assumes that the compressor’s suction pressure is fixed. However, in the real world, this is rarely the case. As we lower the compressor discharge pressure, the compressor suction pressure will also decrease. This will have the normally beneficial effect of reducing the evaporator pressure and temperature.

Now that the evaporator is cooler, we could increase the flow of warm process fluid to the evaporator. Thus, improving the efficiency of the condenser will increase the capacity of a refrigeration

system to cool. This happens mainly by lowering the compressor discharge pressure rather than producing a colder condensed refrigerant.

31.3.2 Condenser Operation

A closed-loop refrigerant condenser ought to be one of the cleanest services in a process plant. Even seal- or lube-oil leaks affect the evaporator efficiency, rather than the condenser. I have measured rather high (e.g., $140 \text{ Btu}/[(\text{h})(\text{ft}^2)(^\circ\text{F})]$) heat-transfer coefficients in such condensers, even after that condenser has been in service for several years since its last cleaning.

The most common problem with refrigeration condensers is condensate backup, rather than fouling (see Chap. 18). This may happen if the condenser outlet line is undersized. Let's say that the pressure drop due to piping friction losses in this line is 10 psi. The refrigerant is isobutane. The isobutane cannot flash as it enters the refrigerant receiver. The reason it cannot partially vaporize is that there is no vapor outlet from the receiver. If it were to vaporize, it would slightly increase the receiver pressure. This would push up the liquid level in the condenser. Many of the tubes in the condenser would be submerged in the liquid refrigerant. The refrigerant would be subcooled below its condensation or bubble-point temperature. It would have to be subcooled, so that after the liquid lost 10 psi of pressure it would not flash.

To subcool isobutane so as to prevent it from vaporizing after losing 10 psi, the isobutane vapor pressure must be reduced by 10 psi. This would, in turn, require cooling the isobutane by about 5°F (see Fig. 14.1 in Chap. 14). Unfortunately, subcooling a liquid is difficult. This means that the heat-transfer coefficient for subcooling might be $30 \text{ Btu}/[(\text{h})(\text{ft}^2)(^\circ\text{F})]$. Even though the amount of heat to be removed in subcooling a refrigerant is small, it may require 30 percent of the tubes in the condenser to do the job. The reason is the low heat-transfer coefficient for subcooling as compared to the high heat-transfer coefficient for condensation.

The tubes in the condenser required for subcooling reduce the heat-transfer surface area required for condensation. In effect, the condenser shrinks. This makes it more difficult to liquefy the refrigerant vapor. The vapor is then forced to condense at a higher temperature and pressure. Of course, this raises the compressor discharge pressure. And, as we have seen in the previous section (Sec. 31.2), this increase in compressor discharge pressure invariably reduces the compressor's capacity and may also increase the horsepower needed to drive the compressor.

An increase in the elevation of the receiver will reduce the pressure of the liquid refrigerant. This loss in pressure due to increased elevation will also require an increase in the surface area of the condenser devoted to subcooling.

31.4 Refrigerant Composition

A very large percentage of refrigeration systems are not limited by the compressor driver horsepower, the condenser, or the evaporator. Many refrigeration systems are limited by either:

- Compressor suction volume limits
- The mechanical pressure rating of the compressor case or the refrigerant condenser shell

If the compressor is suction-volume-limited, we say that it is *speed-limited*. If the compressor is driven by an AC motor, its speed is determined by the motor speed and any connecting gears. If the compressor is driven by a turbine, then the rated speed of the turbine limits the compressor speed.

31.4.1 Overcoming Speed Limits

While the actual cubic feet of refrigerant vapor to the compressor may be limited by the compressor's speed, we can still overcome this bottleneck. The trick is to make the refrigerant lighter or more volatile. A lighter refrigerant will allow the evaporator, shown in Fig. 31.1, to operate at a higher pressure and/or a lower temperature. Increasing the suction pressure increases the density of the refrigerant vapor flowing to the compressor. Even though the volume of gas flow is constant, the weight flow of the circulating refrigerant can be increased. Yes, we can get 6 lb of manure in a 5-lb bag, if we make the manure denser.

For example, adding lighter propane to heavier isobutane in a refinery alkylation unit is a common practice to increase refrigerant circulation. But this will work only when the compressor's driver has spare capacity.

Of course, in process operations, there are no free lunches. Spiking propane into isobutane also makes the lighter refrigerant more difficult to condense. This will raise the compressor discharge pressure. Also, the operating pressure on the shell side of the condenser will increase. If either piece of equipment does not have an adequate design pressure, then increasing the volatility of the refrigerant cannot be permitted.

I used this trick at the Amoco Texas City Refinery in 1975 to increase the refrigeration capacity of my sulfuric acid alkylation unit. When I was put in charge of this alky unit in 1974, I found that capacity was limited to 18,000 BSD by the refrigerant circulation rate. I had plenty of compressor capacity, but as I increased the turbine speed to run my compressor faster and circulate more isobutane refrigerant, the compressor suction pressure declined to 1 psig.

Running the compressor suction below atmospheric pressure is possible, but dangerous. Oxygen can be drawn into the refrigerant loop due to minor air leaks in the suction piping. To avoid this

potential hazard, one should always maintain a positive pressure at the suction of any refrigeration compressor.

To overcome this limitation, I increased the propane content of my circulating isobutane refrigerant from 5 to 10 percent. This required several hours. As the suction pressure rose due to the more volatile refrigerant, the operators increased the refrigerant compressor speed and hence the refrigerant circulation rate.

Eventually, using this and other tricks, I increased unit capacity to 25,000 BSD in 1976. Amoco Oil did not promote me as a consequence of my heroic success. But I remember, and that's what really matters.

31.4.2 Pressure Rating Limits

The simplest way to lower the compressor discharge pressure is to reduce the cooling water temperature to the condenser. One trick is to use the cooling tower makeup water on a once-through basis to the refrigerant condensers. This supply of water is almost always 10 to 20°F cooler than the cooling water supply.

Another way to reduce the compressor discharge pressure is to make the refrigerant heavier or less volatile. Adding isobutane to a propane refrigerant is one example. This will lower the suction pressure to the refrigerant compressor shown in Fig. 31.1 and increase the actual volume of vapor flow. If the compressor is speed-limited, this will not be practical.

Adjusting the refrigerant composition does not significantly affect the amount of horsepower needed to provide a fixed amount of cooling at a certain evaporator temperature. This statement is true even if the type of refrigerant used is changed completely. Propane, isobutane, freon, and ammonia all have roughly the same refrigeration efficiency.

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CHAPTER 32

Cooling Water Systems

I don't know how ordinary people, lacking an engineering background, take care of their swimming pools. Unless you have lots of time, money, and a degree in water chemistry, do not buy a home with a pool.

One thing my pool and process cooling water towers have in common is sludge. Perhaps you have driven past a refinery cooling tower with frothy, white foam billowing from its top. That's caused by the following sequence of events:

1. A tube leak develops in a process condenser.
2. The process-side hydrocarbons leak into the cooling water return line.
3. The hydrocarbons in the circulating warm cooling water (70 to 100°F) promote the formation of algae.
4. The algae die, settle out on the cooling water distribution decks, and form a sludge. The sludge also fouls heat exchanger surfaces and the cooling water distribution piping.
5. The cooling tower is now shocked with chlorine to establish a chlorine residual of a few ppm. If the cooling tower is really dirty, this takes lots of chlorine.
6. The dead and dying algae mix with the air flowing through the cooling tower to form froth.

To maintain a clean cooling water system requires the maintenance of a chlorine residual content of a few ppm. Organic material leaking into the cooling water consumes chlorine. Without a residual chlorine content an organic sludge will form. Thus, the prerequisite for maintaining good heat transfers in cooling water exchangers is to eliminate hydrocarbon leaks into the circulating water system.

32.1 Locating Exchanger Tube Leaks

Open the high point vent on the top of the channel head. I am assuming water is on the tube side of a horizontal shell and tube exchanger. If the exchanger is elevated by 30 or 50 ft, there may be no water pressure at this elevation. If so, temporarily close the cooling water outlet gate valve part way to push the water out of the vent. Now take the gas test meter (the one used by the operators to issue entry permits to vessels) and hold the sample probe above the water stream to test for hydrocarbons.

Do not just plug the leaking tube and start back up. You need to determine the mechanism of the failure. The bad tube has to be extracted from the tube bundle. This is a somewhat difficult job usually requiring an outside contractor.

When you inspect the failed tube, it may well appear that 99 percent of the metal loss is inside the tube on the cooling water side. This suggests that the failure is associated with water-side corrosion. However, if the metal loss is very localized, the problem is on the shell side. Process fluid has jetted through a pinhole leak. The resulting high localized velocity on the tube side creates areas of low pressure inside the tube. Oxygen dissolved in the cooling water can flash out of solution and corrode the surrounding surface of the tube. The problem I'm describing is called *erosion-corrosion* promoted by cavitation. Unless there are signs of general corrosion (thinning, pitting) inside the tubes, large localized tube leaks are due to shell-side, process fluid problems. You cannot draw any conclusions unless the leaking tube is first extracted from the tube bundle.

32.2 Tube-Side Fouling

Sea water fouls rapidly above 130°F. Circulating cooling tower water may cause excessive rates of salt deposits at temperatures above 120°F, depending on the cooling water quality. One common cooling water hardness deposit is calcium carbonate. Just as with a boiler, we need to blowdown some percentage of the circulating cooling water to control the total dissolved solid content of the water. The rate of blowdown is not adjusted based on the water circulation rate, but on the rate of water evaporation from the cooling tower. In many plants the required blowdown rate to meet the TDS content of the cooling water is zero. That's because of leaks and losses in the circulating system.¹

A large percentage of fouling deposits on the cooling water side of exchangers are manganese. This is mainly a result of biological activity, rather than contamination of the cooling water system makeup water. Effective chlorination will suppress such fouling with manganese salts.

One of the mistakes I have made in the past is due to air accumulation in the channel head (see Fig. 32.1). That is, I've confused the effect of trapped air with fouling. Especially on start-up, air may be trapped both above and below the pass partition baffles. The air can fill some of

the tubes, especially at lower tube velocities, and effectively reduce the exchanger surface area. Opening the vents on the channel head to vent out accumulated air is a good method to restore lost cooling capacity. Initially, I had confused this lost cooling capacity with fouling.

32.3 Changing Tube-Side Passes

Once my client in South Africa changed a water cooler from two-pass tubes to four-pass tubes (see Fig. 32.1), meaning that the water traveled through half the number of tubes per pass. The water traveled twice as far, as it now went through the tube bundle four times.

The objective of this change was to increase the water tube-side velocity. Higher velocity means less fouling. But higher velocities are not possible, especially if the water has to travel further, unless the ΔP also increases. Most unfortunately, the ΔP on water-cooled exchangers cannot increase. It must remain constant regardless of the exchanger configuration. Let me explain.

Let's say your sea-water pumps develop a 40-psig discharge pressure. The sea-water return line operates at 5 psig. The pressure drop

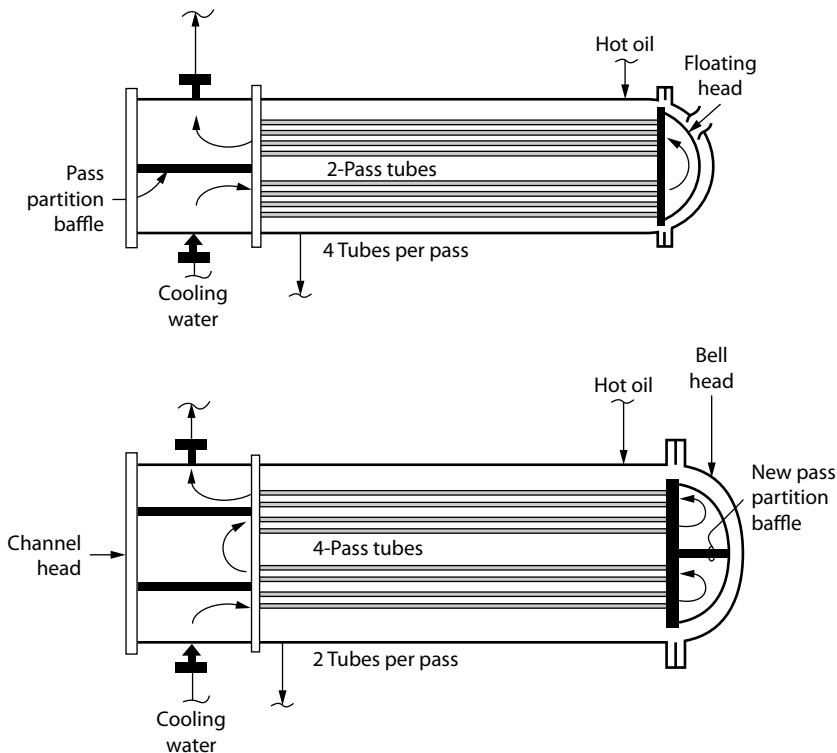


FIGURE 32.1 Converting two-pass to four-pass reduces water velocity at constant ΔP .

of sea water flowing through the exchanger is 35 psig, regardless of the exchanger geometry. If the water flow is restricted by increasing the number of tube-side passes, then both total water flow and the tube-side velocity will decrease.

When my client in South Africa changed from two-pass to four-pass, the water flow decreased from 1,000,000 lb/hr to 350,000 lb/hr. The method of calculation to derive the lower flow is somewhat complex. The calculations take into account that the water has to flow twice as far through the tube bundle, and that there are only half the number of tubes available per pass, but that the water pressure drop must remain constant.

Due to the lower flow, the temperature increase of the sea water also went up:

- Two-pass: 70 to 100°F
- Four-pass: 70 to 150°F

At 150°F, the cooling water precipitated hardness deposits. The hardness deposits restricted water flow. The restricted water flow increased the water outlet temperature. The increased water temperature reduced the water flow. The reduced water flow . . . but perhaps I have made my point. We changed back to the two-pass configuration. Kindly compare this story to Chap. 26, “Shell-and-Tube Heat Exchangers” where tube passes were increased.

32.4 Cooling Tower pH Control

Effective chlorination of circulating cooling water requires proper pH control. Maintaining proper alkalinity also aids pH control and thus also improves the performance of chlorine in suppressing biological activity. A normal pH target is 6.5 to 7.0. If the pH rises well above this target, the cooling water basin becomes brown and murky. I guess that’s because the iron salts dissolved in the water precipitate out of solution. What I know for sure is that heat exchangers throughout the Amoco Refinery in Texas City fouled rapidly due to my operators accidentally overinjecting caustic into the cooling-water return line.

They quickly restored the pH with sulfuric acid. My operators over-shot the target and drove the pH down to 4. The water became clear, clean, and corrosive. Leaks sprang out throughout the carbon steel water supply and return piping. Sulfuric acid is used because the pH of water naturally increases due to evaporation of water in closed systems.

32.5 Wooden Cooling Towers

Most older cooling towers in the United States are constructed of wood. The interior of the tower is filled with wooden slates. The lignite in these wooden slates is removed by the circulating water and

causes the slates to break. The failure of the slates degrades air-water contacting, which raises the supply water temperature to the process units. The two things that will accelerate the rate of lignite loss from the wooden components of the cooling tower are:

- High cooling water return temperature. Above 130°F is considered excessive.
- High residual chlorine levels. Above 2 or 3 ppm is considered excessive. Occasionally shocking the tower with chlorine is common practice, but should be done less than once per week.

32.6 Back-Flushing and Air Rumbling

The operators in the Esso Lago Refinery in Aruba used to blow nitrogen into the water inlet line to coolers. It's safer to use nitrogen than air, in case there is a tube leak. They just coupled a 1-inch hose to the water inlet nozzle and blew in nitrogen for 10 minutes every day. I call this "air rumbling." It helps to clean the tubes.

Back-flushing is more effective in clearing tubes than air rumbling. The cooling water supply line is closed. A valve draining the water to the sewer is opened on the inlet side of the exchanger. The drain, or back-flush valve, has to be at least half the diameter of the water supply piping. Water flows backward through the tubes until the water flowing to the sewer is clear. If the exchanger is elevated more than 20 or 30 ft, it's probable that back-flushing is not possible, due to the lack of water pressure at the higher elevation. In Aruba, I installed a back-flush header pipe using high pressure firewater to supply pressure for back-flushing. Check with your plant's safety division before you make such a modification.

Back-flushing is much more effective when it is done once a week, not as required. It prevents individual tubes from overheating due to low water flow and plugging off with hardness deposits. Once tubes are plugged with carbonate hardness deposits, the next step is acid cleaning, as such tubes cannot be cleared by back-flushing.

32.7 Acid Cleaning

It's best to avoid acid cleaning. pH excursions in the cooling water circulating system are probable. Acid is injected into the water inlet, while a caustic neutralizing agent is injected into the water outlet. Acid cleaning is very effective for the exchanger being cleaned. However, the rest of the cooling water system is subjected to accelerated rates of fouling and low pH corrosion, due to acid escaping from the exchanger this is being acid cleaned. Combating hydrocarbon leaks and improper water chemistry should minimize the need for on-line acid cleaning.

32.8 Increasing Water Flow

Your cooling tower basin is likely served by a number of circulation supply pumps. You may have observed that putting an additional pump on line does very little in increasing the combined pump discharge pressure. This means that the total water flow has not increased. For example:

- Three pumps on: discharge = 48 psig
- Four pumps on: discharge = 50 psig
- Cooling-water return at grade for both cases = 10 psig

The ΔP for the three-pump case is 38 psig. The ΔP for the four-pump case is 40 psig. That means the pressure drop through the exchangers and piping system has increased by 2 psi, or 5 percent. Since pressure drop varies with flow squared:

$$\Delta P \approx (\text{flow})^2$$

The water flow has only increased by 2.5 percent by starting the fourth pump. This does not usually indicate a pump malfunction. More likely, it is an indication that all four pumps are operating on the flat portion of their pump curve. (see Chap. 34, “Centrifugal Pumps: Fundamentals of Operation.”) To get more water flow, you will need to reduce restrictions in the circulating system.

I was working in a refinery in New Jersey. The problem was lack of effluent cooling on a hydrotreater. I decided to increase cooling water flow through the effluent cooler by reducing the water flow through exchangers throughout the rest of the refinery. I reasoned that there was a single cooling tower that served the entire refinery. All the cooling water exchangers were piped-up in parallel. Thus, if I reduced water flow through some of these exchangers, I would divert more water to the hydrotreater effluent cooler.

I monitored progress by looking at both the process and water effluent temperatures on the hydrotreater cooler.

I throttled back on the cooling water flow to dozens of exchangers. Several of these water coolers had not been used for years, but no one had bothered to block-in the water flow. Liz and I spent most of an afternoon on this activity. Based on the combined water temperature returning to the cooling tower (see Fig. 32.3), I had reduced the total circulating water flow by 20 percent. But the water flow to the hydrotreater effluent cooler had not increased at all.

Why? Because I was running on the flat portion of the cooling-water supply pump performance curve.

Was there any reward for my efforts? Yes. The amperage load on the cooling water pump motor was reduced from 60 to 50 amps—an improvement that was of no interest to my client.

32.9 Piping Pressure Losses

For water flowing through clean pipes:

$$\Delta P = 0.15 \times V^2 \div (\text{ID})$$

where ΔP = pressure loss per 100 ft of pipe, psi

V = velocity, ft/s

ID = pipe diameter, inch

The idea is to compare the measured and calculated pressure losses in the water system. In sea-water systems, the pipe lining may tear loose and restrict the water flow. I once found an isolation gate valve mostly closed, buried under a road crossing. It was a 30-inch valve. A maintenance crew spent two days working it open. The water flow only increased by 8 percent. The plant manager was quite angry. So the manager initiated a project to install a new 500-hp pump to run in parallel with the existing two 500 hp. As the existing pumps were already running on the flat portion of their performance curves, the incremental water flow observed when the new pump was commissioned was 2 percent.

32.10 Cooling Tower Efficiency

Two types of cooling towers are in widespread use:

- Natural draft (see Fig. 32.2)
- Induced draft (see Fig. 32.3)

The giant natural-draft towers are used in Europe and in nuclear power plants in the United States. I've only worked with these towers in one refinery in Lithuania, and their performance was poor. Air flow is generated by the air inside the tower being heated by the warm cooling water. Also, the molecular weight of the evaporated water is less than the molecular weight of air. A few tenths of an inch of water draft may be developed.

The much smaller induced-draft cooling towers are used in process plants throughout the Gulf Coast of Texas and Louisiana. Air flow is generated by large induced-draft fans.

Regardless of the type of cooling tower, the same criterion is used to gauge performance: the approach of the cooled water temperature compared to the air wet bulb temperature.

32.11 Wet Bulb Temperature

Water is not significantly cooled by exchanging sensible heat with cold air. Most of the cooling results from the humidification of the air. If you live in the Central Valley of California, your home is likely

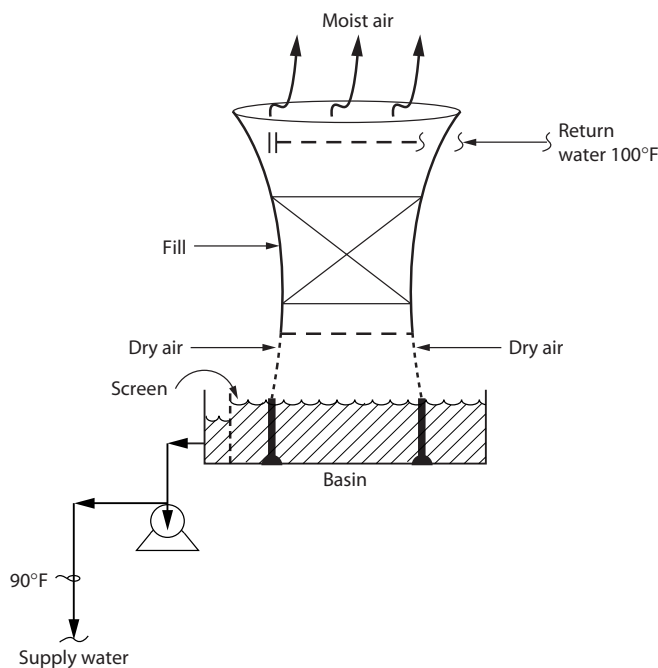


FIGURE 32.2 Natural-draft cooling tower.

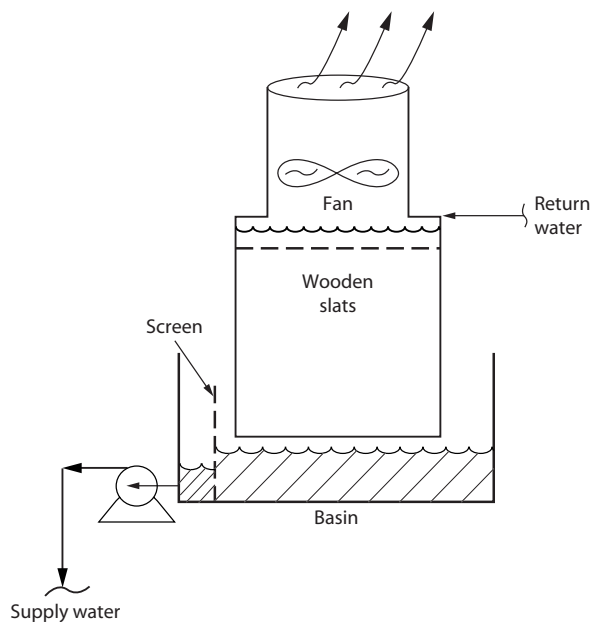


FIGURE 32.3 Induced-draft cooling tower.

cooled with a “swamp cooler,” rather than with a freon compressor. The water partially evaporates as it mixes with the dry air. The latent heat of evaporation of water is 1000 Btu/lb. If 2 percent of the water evaporates by contact with the cold air, the water loses 20 Btu/lb. The specific heat of water is one Btu/lb per 1°F. Thus, the water will be cooled by 20°F by evaporation.

It's my mother's mushroom soup story again (Sec. 1.4). Converting the sensible-heat content of the water into latent heat of evaporation of the water. But for the air to carry away the evolved water vapor, it has to be below its dew-point temperature.

To gauge how efficiently a cooling tower is working requires two temperatures:

- The cooling water supply temperature leaving the basin.
- The wet bulb temperature of the ambient air. The air temperature itself is called the dry bulb temperature, which is not important.

To get the wet bulb temperature, turn on your TV. Switch to the weather channel. Read the current dew-point temperature, which is the same as the wet bulb temperature.

Alternatively, measure the wet bulb temperature as follows: Take a small piece of fabric and loosely wrap the bulb of a thermometer, leaving a short tail of the fabric dangling below the thermometer. Dip the tail of fabric into water so that the water soaks all the way up the fabric that is covering the thermometer bulb. Now measure the ambient air temperature using the thermometer with the fabric wrapped around its bulb. This is known as the wet bulb temperature. It will be lower than the dry bulb or ambient air temperature because the water vaporizing off the fabric cools the bulb of the thermometer.

Calculate the performance ΔT . This is the difference between the water leaving the tower and the wet bulb temperature. You can judge the performance of the cooling tower as follows:

- If the performance ΔT is 5°F or less and the temperature rise of the water (the return minus the supply) is 15°F or more, tower performance is excellent.
- If the performance ΔT is 8°F or less and the temperature rise of the water (return minus supply) is 30°F or more, the tower performance is also excellent.
- If the performance ΔT is 10°F and the warm water coming back to the cooling tower has increased in temperature by 15°F or less, performance is poor. Note that the 15°F is called the water temperature rise.
- If the performance ΔT is 15°F and temperature rise of the water is 30°F, tower performance is poor.

In the older, induced-draft wooden cooling towers, a common problem is that the wooden slates have broken up. There is usually an access inspection door at the base of the cell. Often, the water distribution holes are plugged or there are large holes broken into the deck. Water is not supposed to overflow around the outside of the cooling tower. On some cooling towers the fans are driven by belts, which can slip or the fan blade pitch is set too low, low being less than 15°, maximum being 23°.

So here's another true story. April 1974, Texas City. I noted that the holes on my cooling tower distribution decks were blocked with a slimy sludge. I shut off the water to one cell and hired a crew of laborers to clean the deck. I did not think to provide safety harnesses or belts. Naturally, one of those guys fell off and broke his arm. Frank Citek, my boss, asked if I was crazy or stupid or both.

Actually, I was just ignorant, which is a curable deficiency. Next time, I first eliminated hydrocarbon leaks into the cooling water system and chlorine shocked the cells, using bottled chlorine.

But this was even worse. Julia Tieneman, a young and inexperienced operator in Texas City in 1974, was assigned to my unit. Why my foreman, old Henry Zipreian, chose Julia to connect the chlorine bottles to the water slip stream, I can't say. Dear Julia got a whiff of chlorine and went off on temporary disability pay, which I charitably approved.

Rather than stay home and recuperate, Julia went on vacation to Mexico. Most unfortunately, she ran into Ralph Samuels, the refinery human resources manager. Mr. Samuels complained to Frank Citek, my boss, "Why was Julia getting disability pay to cavort in Tijuana?"

"Because," Mr. Citek explained, "Lieberman is crazy and stupid."

Reference

1. R. E. Putman, *Steam Surface Condensers* (ASME Press, 2001).

CHAPTER 33

Catalytic Effects: Equilibrium and Kinetics

When I discuss a process problem, I'll frequently qualify my views by stating, "Let's assume equilibrium conditions have been reached." What I'm doing is choosing to ignore the effect of time on changes in composition. I'm ignoring the effects of kinetics on the process. I'll make such assumptions to simplify complex discussions. It's a bad habit, which sometimes gets me into trouble.

33.1 Kinetics vs. Equilibrium

A reaction that is limited by equilibrium is finished. Waiting longer will make no difference. An example of this is the absorption of H_2S from refinery fuel gas into an amine solution. Depending on the concentration of the H_2S in the fuel gas, the amine solution will become saturated with H_2S . Mixing the fuel gas and the amine solution for a longer period of time will *not* push more of the H_2S into the amine solution. The reaction is complete, and we say the absorption of H_2S by amine is limited by equilibrium.

On the other hand, a reaction that is limited by kinetics is *not* finished. Waiting longer will help to complete the reaction. An example of this is the absorption of CO_2 from refinery fuel gas into an amine solution. The CO_2 will react with the amine to produce a stronger chemical bond than would the H_2S and amine. CO_2 reaction with amine goes slower than the H_2S reaction with amine. Mixing the fuel gas and the amine solution for a longer period of time will push more of the CO_2 into the amine solution. The reaction is incomplete, and we say the absorption of CO_2 by amine is limited by kinetics or time.

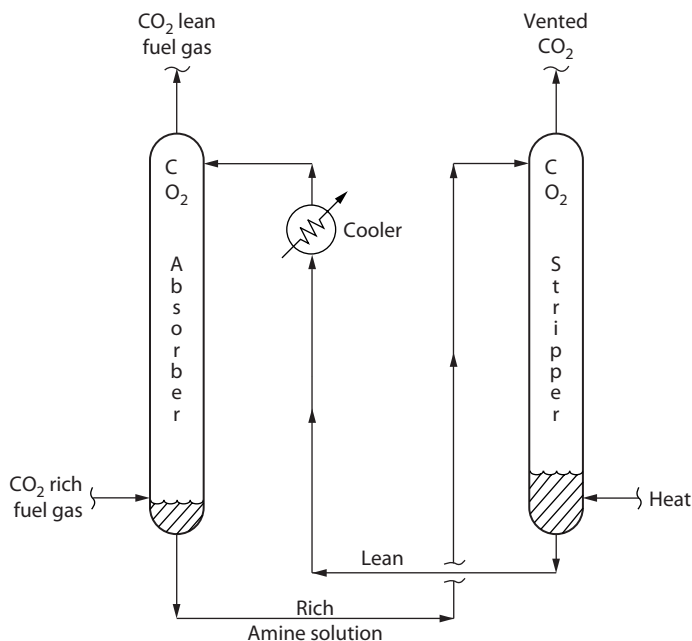


FIGURE 33.1 Recovery of CO₂ with amine solution.

In the plant, then, we have two problems:

- H₂S absorption is limited by equilibrium.
- CO₂ absorption is limited by time.

We deal with these two problems by:

- To extract the residual H₂S from fuel gas, we circulate more amine solution.
- To extract the residual CO₂ from fuel gas, we have more contacting trays in the absorption tower.

A similar problem occurs in the amine solution regenerator or stripper (see Fig. 33.1). The H₂S strips out easily. Residual CO₂, which is more tightly bound to the amine molecule, requires lots of trays and lots of steam to strip out.

33.2 Temperature vs. Time

If a reaction is too slow, there is a simple way to make the reaction go faster: higher temperature. Milk goes sour even in my 38°F refrigerator after a few months. The milk will go sour much faster if I leave it on our kitchen table at 74°F for a few days. Typically, for a nonbiological

reaction, the rate of reaction might very roughly double for every 30°F increase in temperature.

Regarding the example of CO₂ absorption in amine solution, I said the CO₂ absorption was limited by time or kinetics or reaction rate and not equilibrium. Perhaps I could increase the amount of CO₂ absorption in the amine solution by increasing the absorber temperature shown in Fig. 33.1.

Maybe yes, but maybe no. Maybe yes, because the reaction will go faster with higher temperature. But maybe no, because a higher temperature reduces the amount of CO₂ that, at equilibrium, can be absorbed by the amine solution.

33.3 Purpose of a Catalyst

Increasing temperature to speed up a reaction is often a bad idea. If I have a plant that is desulfurizing diesel oil with hydrogen, so as to convert the sulfur in the diesel to H₂S, I can achieve the required percent of sulfur removal in two ways:

- Mix the hydrogen and diesel together and wait a very long time.
- Mix the hydrogen and diesel together at a very high temperature (850°F).

At 850°F, desulfurization will proceed quickly, but the diesel oil will also thermally crack to coke and gas and be useless as a fuel for trucks. What is wanted is a way to accelerate the conversion of the sulfur molecules attached to the diesel to H₂S at a moderate temperature (about 700°F). That's what a catalyst does. It does not change the equilibrium, it just makes the reaction go faster without having to increase the temperature.

For a catalyst to be effective in accelerating the rate of reaction, it has to come into contact with the molecules of the stuff that's reacting. That's why catalysts have lots of surface area. Most of the surface area of a pellet of catalyst can only be seen with a microscope. As we try to desulfurize diesel oil, some small fraction of the hydrocarbon will turn to coke. The coke covers a tiny portion of the surface area of the catalyst. This slows down the rate of reaction between the hydrogen and sulfur molecules converting to H₂S in the diesel oil. The sulfur content of the diesel product goes above spec. To reduce the sulfur content of the diesel below spec, the operator raises the temperature of the reactor. This makes the reaction go faster, which is good. But it also increases the rate at which coke deposits on the surface of the catalyst, which is bad. The extra coke blinds-off more catalyst area. The reaction rate drops. More sulfur is left in the diesel. The operator increases the reactor temperature to speed up the reaction, which leaves more coke on the catalyst. When this game is over, we are at the "end-of-run condition."

33.4 Lessons from Lithuania

Sometimes my habit of assuming equilibrium gets me into trouble. A few years ago I was hired by a refinery in Lithuania to redesign their hydrogen production plant, to increase plant capacity by 25 percent, and to reduce the CO_2 content of the hydrogen product. The CO_2 content of the hydrogen was 2 percent, and the new specification was 0.5 percent CO_2 in hydrogen product.

The process for CO_2 absorption is the same as shown in Fig. 33.1. The only difference is that a potassium salt solution is used instead of the amine solution to absorb the CO_2 from the hydrogen product. The potassium salt solution is called Catacarb, as shown in Fig. 33.2.

To accelerate the rate of absorption and deabsorption of CO_2 , a catalyst called LRS-10 is added to the Catacarb. I had ordered a fresh supply of LRS-10 catalyst for start-up. When I arrived in Lithuania, I was informed that the LRS-10 catalyst was stuck in customs and would be delivered any day now. I decided not to worry about the LRS-10 catalyst for two reasons:

1. The plant operations chief informed me that there was already a catalyst in the Catacarb solution that would be used for start-up. It was a di-ethanol amine catalyst, not LRS-10, but still it was a catalyst.
2. You can't worry about everything.

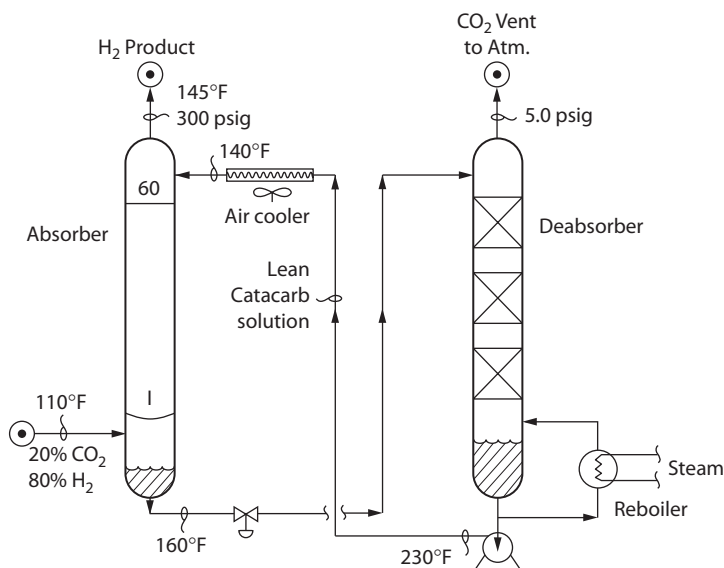


FIGURE 33.2 Circulating Catacarb solution absorbs CO_2 from hydrogen.

As lower temperatures favor the absorption of CO_2 by the Catacarb solution, I had expanded the capacity of the circulating Catacarb solution aerial fin-fan cooler to cool the absorber tower and thus reduce CO_2 in the H_2 product. Liz stood next to me at the panel when we started the Catacarb circulation pumps. I was totally confident.

I forgot to mention that the operators in Lithuania didn't like me. Nothing personal. It was just that they didn't like Americans. So when the cooled Catacarb solution began to circulate, they were overjoyed. The CO_2 content of the hydrogen product lined out at 3.5 percent CO_2 , well above the previous level of 2 percent, and far above my target of 0.5 percent CO_2 in the hydrogen product.

Liz said that I went deathly pale. I was soaked with sweat in the chilly control room. The operations chief and panel room operators beamed with pleasure. Confused, I asked them to turn on the last fan on the aerial fin-fan solution cooler. The absorber temperature dropped by 10°F and the CO_2 in the hydrogen product went up from 3.5 to 4 percent CO_2 .

"Any further instructions, Comrade Lieberman?" the unit chief happily inquired. I didn't answer. I couldn't breathe.

For the next three weeks I engaged in a titanic battle with the hydrogen plant. Fourteen hours a day, seven days a week:

- I raised the absorber pressure by 15 psi, which helped a bit.
- I lowered the deabsorber pressure by 3 psi, which helped a bit.
- I maximized the steam flow to the deabsorber reboiler, which helped a bit.
- I had open stripping steam piped up to the reboiler return nozzle, which helped a bit.
- I increased the circulation rate of the Catacarb solution, which helped a bit.
- I shut off one of the four aerial cooler fans on the lean Catacarb solution (see Fig. 33.2), which helped a lot.

So I shut off, one by one, over a period of days, all the aerial cooler fans. The absorber became quite hot and the CO_2 dropped to 0.5 percent in the H_2 product. None of this made any sense. CO_2 absorption in Catacarb solution should be favored by lower, not higher, temperatures. Equilibrium conditions for CO_2 absorption are always improved by cooling the circulating absorption liquid.

33.4.1 The Celebration Dinner

The general director of the plant, Nelson English (an American), invited Liz and me to a dinner to celebrate the 0.5 percent CO_2 . Waiting for the main course, I thought about the problem. Why would raising the absorber temperature, clearly bad for equilibrium, reduce

CO₂ in hydrogen? Certainly absorption efficiency is improving, not because of equilibrium, but because of kinetics. It must be reaction rate or time limited. Otherwise, the higher temperatures in the absorber would never help.

"Mr. English," I asked, "what about the LRS-10 catalyst? Has it cleared customs in Klipiada yet (the old German port of Memel)?"

33.4.2 Catalytic Effect

The catalyst was like a cold drink on a hot summer's day. I put all the variables I had changed in three weeks back to normal conditions. I turned on all the aerial cooler fin-fans. The CO₂ dropped below 0.2 percent in the hydrogen product.

The lesson is, if raising reaction temperature in a reactor promoting an exothermic reaction makes the reaction proceed further, the reactor is limited by kinetics and not equilibrium. The term "exothermic" means that heat is liberated in the reaction. How can one tell if the absorption of CO₂ in the Catacarb solution is exothermic (gives off heat), rather than endothermic (absorbs heat)? Refer to Fig. 33.2. Note that both the hydrogen gas and the Catacarb liquid are being heated in the absorber. They are being heated by the heat of reaction between the CO₂ and the potassium salt in the Catacarb solution.

33.5 Zero Order Reactions

Not all reactions are exothermic. Thermal cracking is an endothermic reaction. Heat is absorbed. Good thing, too. If thermal cracking of crude oil was exothermic, all the earth's crude would by now have turned to coal and natural gas. Delayed cokers, visbreakers, and fluid catalytic cracking units are processes that are primarily endothermic in nature. A delayed coker operates with a zero order reaction. This means the rate of reaction depends on time in the coke drum and the temperature in the coke drum. The composition of the products of reaction have no effect.

On the other hand, hydrocracking is exothermic. The consumption of hydrogen liberates heat in the hydrocracking reaction.

33.6 Runaway Reaction

In 1968, a 3000-psig vessel exploded. This unhappy event occurred at the ESSO Refinery in Bayway, New Jersey. The cause was a runaway reaction. Tar was being hydrocracked in an emulated bed of catalyst. The reactor got hotter, more hydrogen was consumed, and more exothermic heat of reaction was generated, which increased the rate of hydrogen consumption, which released more exothermic heat of reaction. I suppose if hydrogen feed had been stopped sooner, the runaway reaction could have been controlled. But it wasn't and the reactor blew apart.

I had a similar, but less dramatic experience on a liquid-filled reactor converting propylene to viscous polypropylene (a gasoline additive). The catalyst was aluminum chloride added continuously with a small pump. The pump malfunctioned. During its outage we failed to stop feed to the reactor. The propylene content of the reactor increased. Finally, we restored the catalyst flow, but the liquid-filled reactor became very warm, very fast. Fortunately, an alert operator stopped the propylene feed to the reactor in time.

33.7 Common Chemical Plant and Refinery Catalytic Processes

33.7.1 Naphtha Reforming

This is an endothermic reaction in a hydrogen environment taking place in the vapor phase. The catalyst is the size of grains of rice and contains very costly platinum (which sells for twice the price of gold). The objective is to produce aromatics benzene, toluene, or xylene. Benzene and xylene are the basic building blocks of the chemical industry.

33.7.2 Steam-Hydrocarbon Reforming

This is also an endothermic process. The reaction takes place inside catalyst-filled furnace tubes. Heat is supplied to the tubes by furnace firing. Almost any hydrocarbon can be used—natural gas, propane, butane. The objective is to make hydrogen. Most of the world's hydrogen is produced in this way. Half the hydrogen comes from steam and the other half from hydrocarbons. All of the carbon in the feed is converted to CO_2 and vented. How hydrogen-powered cars will help global warming, I do not grasp.

33.7.3 Alkylation

This is a very exothermic reaction. Iso-butane and light olefines react to form iso-octane, a gasoline blending component with a 100 octane. That's where the word "octane" comes from. Reaction takes place at 50°F , in a liquid phase of either H_2SO_4 or HF acid.

33.7.4 Polymerization

This is an old process using a solid bed of phosphoric acid. Reaction takes place in the vapor phase. Feed is normally propylene. Products are hexane and nonane. Polymerization is a very exothermic process.

There is another sort of polymerization process used to produce viscous polypropylene. The reactor is a refrigerated kettle evaporator where the reaction takes place in the liquid phase. The catalyst was iso-propyl-chloride. The polypropylene product was used by Texaco in the 1960s as an engine top cylinder lubricant, added to gasoline.

One c.c. per gallon. The polypropylene was made at the Amoco refinery in El Dorado, Arkansas. I produced several designs to expand plant capacity and participated in the plant start-up after each of my design changes was implemented.

Texaco and Amoco are gone. The refinery in El Dorado has vanished. All that remains is a picture on my wall of a young man in a round hard hat, leaning against the old poly reactor.

33.7.5 Sweetening

This is a slightly exothermic process. Mercaptans are converted to disulfides by adding air. The catalyst is a blue powder (Merox), which impregnates a bed of charcoal saturated with caustic. Converting mercaptans to disulfides reduces the bad smell of gasoline and jet fuel. The process takes place at about 100°F. Sulfur content of the product is not changed.

33.7.6 White Oil Hydrogenation

Motor oil is hydrotreated at 3000 psig and 450 to 650°F to remove all aromatic compounds by contact with hydrogen. The resulting white oil is used for baby oil, mineral oil, and ladies' cosmetics. The white oil costs \$5 per gallon to make and is sold for \$30 an ounce in face creams.

33.7.7 Fluid Catalytic Cracking

Reaction takes place at 1000°F and low pressure in a riser pipe. The catalyst looks like white sand but is actually made of zeolites. The catalyst picks up coke from the endothermic cracking reaction. The coke is burned off of the circulating catalyst with air. The catalyst gets hot (1200°F) and this supplies the endothermic heat of reaction. Half the gasoline sold in the United States is produced in this way. The original catalyst for this process was actually beach sand.

33.7.8 Sulfuric Acid

Sulfuric acid is produced by burning sulfur and H_2S to SO_2 with air. Then more air is added and reacted in a catalyst bed of vanadium pentoxide to produce SO_3 . The SO_3 is mixed with water to make sulfuric acid. The reaction is exothermic and takes place in the vapor phase. Sulfuric acid is, by weight, one of the main chemicals produced in the world. I spent 1974 to 1976 in charge of a sulfuric acid regeneration plant in Texas City. In 1975, I had 34 shutdowns and start-ups. I've kept a piece of lead wire, part of that awful plant, in my desk drawer. When things seem bad, that memento reminds me that it could be a lot worse.

CHAPTER 34

Centrifugal Pumps: Fundamentals of Operation

Head, Flow, and Pressure

Most of the pumps we see, both at home and at work, are centrifugal pumps. And most of these pumps are driven by constant-speed, alternating-current motors.

34.1 Head

Centrifugal pumps are *dynamic machines*, which means that they convert velocity into feet of head.

To explain this concept of converting speed or velocity into feet of head, let's look at Fig. 34.1. This is the water intake section of the Chicago Water Treatment Plant. Water flows from Lake Michigan into a large concrete sump. The top of this sump is well above the level of Lake Michigan. The line feeding the sump extends three miles (3 mi) out into the lake. The long line is needed to draw water into the plant, away from the pollution along the Chicago lakefront. The pipeline diameter is 12 ft. Water flows in this line at a velocity of 8 ft/s. Six pumps, stationed atop the concrete sump, pump water into the water treatment plant holding tanks.

One day, the plant experiences a partial power failure. Three of the six pumps shown in Fig. 34.1 shut down. A few moments later, the manhole covers on top of the sump blow off. Geysers of water spurt out of the manholes. What has happened?

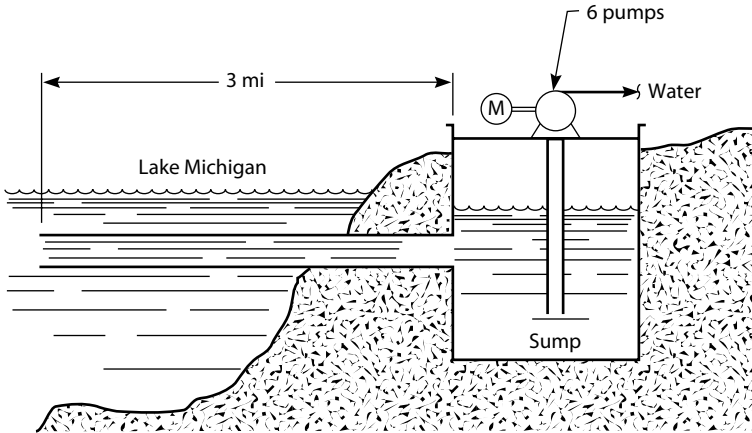


FIGURE 34.1 Converting momentum into feet of head.

34.1.1 Hydraulic Hammer

This is an example of water, or hydraulic, hammer. But what causes water hammer?

Let's consider the change in the water velocity in the 3-mi pipeline when half the pumps failed. The velocity in the pipeline dropped from 8 down to 4 ft/s. *Velocity* is a form of energy, called *kinetic energy*. Energy can take several different forms:

- Heat (Btu)
- Elevation or potential energy
- Kinetic energy (miles per hour)
- Pressure (psig)
- Electrical power (amperes)
- Work (horsepower)
- Acceleration
- Chemical (heat of reaction)

About 250 years ago, Daniel Bernoulli first noticed two important things about energy:

- Energy in one form can be converted to energy in another form.
- While energy can be changed from one form to another, it cannot be created or destroyed.

This idea of the conservation of energy is at the heart of any process plant. In the case of the Chicago Water Treatment Plant, the reduced velocity of the water was converted into feet of head. That is,

the elevation of the water in the sump suddenly increased and blew the manhole covers off the top of the sump.

34.1.2 Momentum

If the length of the pipeline had been a few hundred feet, this incident would not have happened. It is not only the sudden reduction of the velocity of the water that caused an increase of the water level in the sump. It is also the mass of water in the 3-mi pipeline that contributed to the increased height of water in the sump. The combined effect of mass times velocity is called *momentum*.

The mass of water in our pipeline weighed 160×10^6 lb. This much water, moving at 8 ft/s, represents a tremendous amount of energy (about 500 million Btu per hour). If the flow of water is cut in half, then the momentum of the water flowing in the pipeline is also cut in half. This energy cannot simply disappear. It has to go somewhere. The energy is converted to an increase of feet of head in the sump; that is, the water level in the sump jumps up and blows the manhole covers off the top of the sump.

Incidentally, this is a true story. Can you imagine what would have happened if all six pumps failed simultaneously due to an electric power failure? The result would be a dramatic lesson in the meaning of water hammer.

34.1.2.1 Sayano-Shushenskaya Hydro Accident

In 2009, long after I penned the above words, the hypothetical incident I've described above did actually happen, with tragic results. A new hydroelectric plant had just come on stream in Siberia. It was the world's sixth largest. An elaborate celebration tour was being held for the visiting dignitaries. Reportedly, a huge tree stump became lodged in the turbine blades of one of the giant water intake tunnels. The effect of the resulting hydraulic surge may be seen on <http://en.wikipedia.org> and www.youtube.com.

I've printed a few of the hundred odd photos available from the Internet. I use them to frighten the engineers in my trouble-shooting seminars. Seventy-four people were killed in the plant. Check the photos out for yourself. Of course, within days of the disaster, the Russians had modified the story. But one picture is worth a thousand words.

34.1.3 My Washing Machine

Figure 34.2 is a picture of our washing machine as it was originally installed. Whenever the shutoff valve on the water supply line closed, water hammer, or hydraulic shock, would shake the water piping. The momentum of the water flowing in the piping would be suddenly converted to pressure. If the end of a piping system is open (as into a sump), then the momentum of the water is converted to

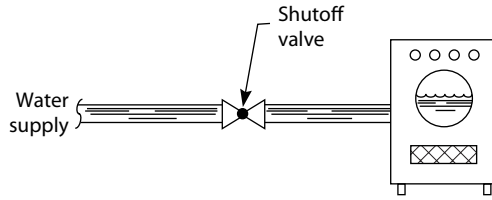


FIGURE 34.2 Hydraulic hammer hits home.

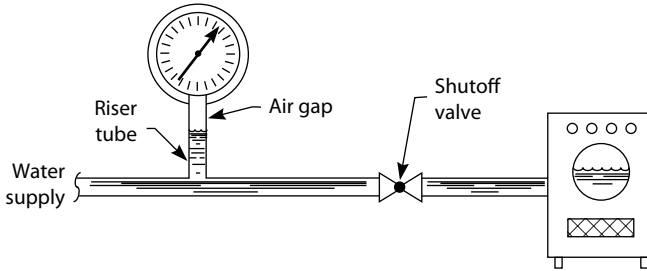


FIGURE 34.3 Riser tube stops hydraulic hammer.

feet of head. But if the end of the piping system is closed, then the momentum of the water is converted to pressure.

To fix this problem, I installed the riser tube shown in Fig. 34.3. The top of the riser tube is left full of air. Now when the water flow in the supply pipe is shut, the momentum of the water is converted to compression energy. That is, the air in the riser tube is slightly compressed, as indicated by the pressure gauge I installed at the top of the riser tube.

I invite the reader to compare my washing machine story to the accident at the Siberian hydroelectric plant. The lesson is that when we have fluid flowing in a very long pipe or channel, we have to consider what will happen when the fluid flow suddenly stops. We do not want the momentum to be converted into pressure, but into compression of a vapor or into potential energy.

For example, on a long crude oil transportation pipeline, we might have periodic surge drums located along the line to safely accommodate a sudden stoppage of the flowing crude.

An easy way to witness hydraulic surge for yourself is to install a pressure gauge at the suction of a centrifugal pump. Then shut down the pump. You will note a brief, but definite increase, in the pump's suction pressure. This is the opposite of the effect I'll now discuss.

34.1.4 Acceleration

Let us imagine that the six pumps in Fig. 34.1 have not run for a few days. The water level in the sump and the level of Lake Michigan will be the same. I now start all six pumps at the same time. An hour later, the water level in the sump is 12 ft below the water level in the lake.

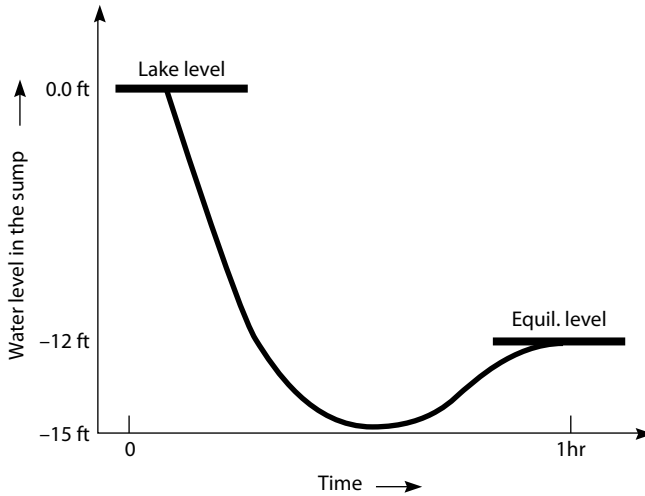


FIGURE 34.4 Effect of acceleration.

This 12 ft is called “feet of head loss.” If the pipeline is 3 mi long, we say we have “lost 4 ft of head per mile of pipeline.”

If we look down into the sump, what would we see happening to the water level during this hour? Figure 34.4 is a graph of what we would observe. The water level in the sump would drop gradually to 15 ft below the level in the lake. Then the water level in the sump would come partly back up to its equilibrium level of 12 ft below the lake level. Why?

The loss of 12 ft of head as the water flows through the pipeline is due to friction; that is, 12 ft of head are converted to heat. But why do we have a temporary loss of an extra 3 ft? The answer lies in the concept of acceleration.

Let’s say you are driving your car onto the expressway. To increase the speed of your car from 30 to 65 miles per hour, you press down on the accelerator pedal. Having reached a velocity of 65 miles per hour, you ease off the accelerator pedal to maintain a constant speed. Why? Well, according to Newton’s second law of motion, it takes more energy to accelerate your car than to keep it in motion.

Referring again to Fig. 34.1, the water in the 3-mi pipeline is initially stagnant. Its velocity is zero. An hour later, the water has accelerated to 8 ft/s. When you accelerate your car, the extra energy required comes from the engine. But when we accelerate the water in the pipeline, where does the extra energy come from?

Does this extra energy come from the pumps? Absolutely not! The pumps are downstream of the pipeline and the sump. They cannot contribute any energy to an upstream pipeline. No, dear reader, the energy to accelerate the water in the pipeline must come from Lake Michigan.

But what is the only source of energy that the lake possesses? Answer—elevation or potential energy. To accelerate the water in the pipeline to 8 ft/s requires more energy than to keep the water flowing at that same velocity. And this extra energy comes from the extra 3 ft of elevation difference between the water level in the sump and the water level in the lake. Once the water has reached its steady-state velocity of 8 ft/s, the need for this extra conversion of feet of head to acceleration disappears, and the water level in the sump rises to within 12 ft of the lake's level.

34.2 Starting NPSH Requirement

The need to accelerate the fluid in the suction of a pump is called the starting *net positive suction head* (NPSH) requirement. To calculate this starting NPSH requirement, let's assume:

- Suction line = 100 ft
- Eight inch suction line
- Fluid = water @ 62 lb/ft³
- Initial velocity is 0
- Final velocity is 10 ft/s

Then proceed as follows:

- Mass in line is 2170 lb
- Change in momentum in line is 10 ft/s times 2170 lb = $22,170 \frac{\text{ft} \cdot \text{lb(m)}}{\text{s}^2}$

If the operator increases the flow rate uniformly over a period of 100 seconds, the change in momentum is

$$22,170 \frac{\text{ft} \cdot \text{lb}}{\text{s}^2} \div 100 \text{ s} = 222 \frac{\text{ft} \cdot \text{lb(m)}}{\text{s}^2}$$

One lb mass equals 1 lb force times s²/ft. Therefore, the force needed to accelerate the water in the pump's suction is

$$222 \text{ lb (f)}$$

The area of the 8-inch suction line is 50 in². Therefore, the starting pressure loss is

$$\begin{aligned} 222 \text{ lb (f)} \div 50 \text{ inch}^2 &= 4.4 \text{ psi} \\ \text{or } 4.4 \text{ psi times } 2.31 &= 10.1 \text{ ft of water} \end{aligned}$$

34.3 Pressure

34.3.1 Loss of Suction Pressure

The need to accelerate liquid in the suction line of a pump leads to a difficult operating problem, which occurs on start-up. Just before the pump shown in Fig. 34.5 is put on line, the velocity in the suction line is zero. The energy to increase the velocity (i.e., accelerate) the liquid in the suction line must come from the pressure of the liquid at the pump's suction. As the pump's discharge valve is opened, the velocity in the suction line increases, reducing the pressure at the suction of the pump. The faster the discharge valve is opened, the greater the acceleration in the suction line, and the greater the loss in the pump's suction pressure.

If the pressure at the suction of the pump falls to its bubble or boiling point, the liquid will start to vaporize. This is called *cavitation*. A cavitating pump will have an erratically low discharge pressure and an erratically low flow. As shown in Fig. 34.5, the bubble-point pressure of the liquid is the pressure in the vessel. We usually assume that the liquid in a drum is in equilibrium with the vapor. The vapor is then said to be at its dew point, while the liquid is said to be at its bubble point.

To avoid pump cavitation on start-up, the experienced operator opens the pump discharge valve slowly. Slowly opening the discharge valve results in reduced acceleration of the liquid in the suction line and a slower rate of the conversion of suction pressure to velocity.

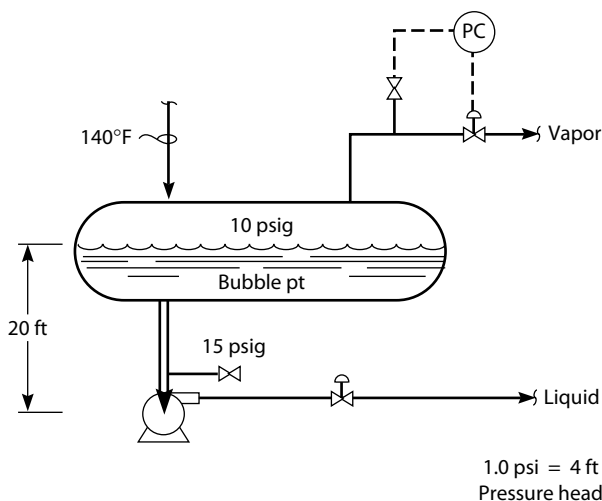


FIGURE 34.5 Loss of suction pressure causes cavitation.

Try this test. To illustrate what I have just explained, try this experiment:

1. Open the suction valve to a pump completely.
2. Crack open the case vent to fill the case with liquid. This is called “priming the pump.”
3. Open the pump discharge valve completely.
4. Push the motor START button.
5. Observe the pump suction pressure.

You will see that the pump suction pressure will drop and then come partly back up. If the pump suction pressure were 15 psig to start with, it might drop to 12 psig and then come up to 14 psig. The permanent difference in the suction pressure between 15 and 14 psig is due to a 1-psi piping friction pressure loss. The temporary difference in the suction pressure between 14 and 12 psig is due to a 2-psi conversion of pressure to velocity or kinetic energy.

To review—all the energy needed to accelerate the liquid to the suction of a pump comes from the pump’s suction pressure. None of this energy comes from the pump itself. Or, as one clever operator at the Unocal Refinery in San Francisco explained to me, “Pumps push, but they do not suck.”

34.3.2 Pump Discharge Pressure

Figure 34.6 illustrates the internal components of an “overhung, single-stage” centrifugal pump. The term “overhung” refers to the feature that the pump has only an inboard, but no outboard, bearing. The inboard side of a pump means the end closest to the driver. The term “single stage” means that there is only one impeller. Multistage pumps can have five or six impellers.

The main components of the pump shown in Fig. 34.6 are

- *Shaft*—used to spin the impeller
- *Coupling*—attaches the shaft to the turbine or motor driver
- *Bearings*—support the shaft
- *Seal*—prevents the liquid inside the pump from leaking out around the shaft
- *Impeller wear ring*—minimizes internal liquid leakage, from the pump discharge, back to the pump suction
- *Impeller*—accelerates the liquid
- *Volute*—converts the velocity imparted to the liquid by the impeller to feet of head

The impeller is the working part of a centrifugal pump. The function of the impeller is to increase the velocity or kinetic energy

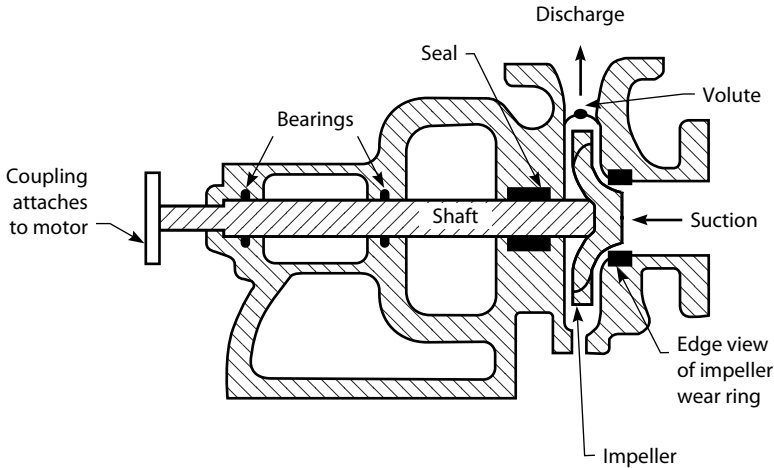


FIGURE 34.6 A centrifugal pump.

of the liquid. The liquid flows into the impeller and leaves the impeller at the same pressure. The location of the oval dot shown at the top of the impeller in Fig. 34.6 is called the “vane tip.” The pressure at the vane tip is the same as the pump’s suction pressure. However, as the high-velocity liquid escapes from the impeller and flows into the volute, its velocity decreases. The volute (which is also called the *diffuser*) is shaped like a cone. It widens out in the manner illustrated in Fig. 34.7. As the liquid flows into the wider section of the volute, its velocity is reduced, and the lost velocity is converted—well, not into pressure, but into feet of head.

34.3.3 Feet of Head

A centrifugal pump develops the same feet of head regardless of the density of the liquid pumped, as long as the flow is constant. This statement is valid as long as the viscosity of the liquid is below 50 to 100 cP or 200 to 400 SSU (Saybolt seconds universal). But, as process operators or engineers, we are not interested in feet of head. We are interested only in pressure. Differential pressure is related to differential feet of head as follows:

$$\Delta P = \frac{(\text{s.g.}) \Delta H}{2.31}$$

where ΔP = increase in pressure, psi

ΔH = increase in feet of head, ft

s.g. = specific gravity of the liquid (i.e., the density of the liquid, relative to cold water)

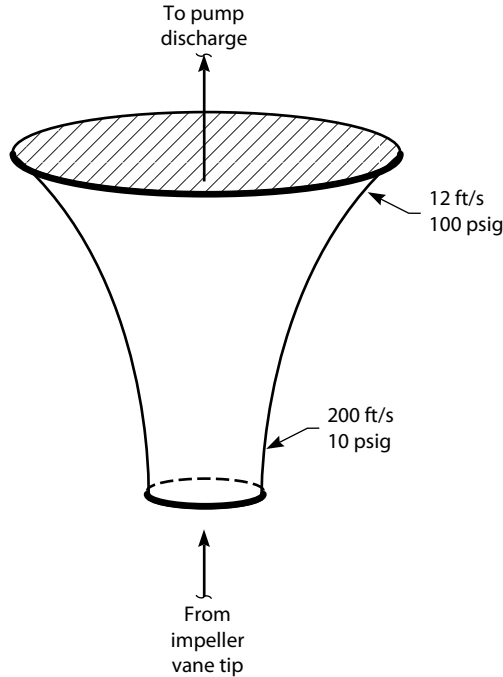


FIGURE 34.7 A volute or diffuser converts velocity into feet of head.

The increase in pressure ΔP is also called *head pressure*. For example, if I have 231 ft of water in the glass cylinder shown in Fig. 34.8, it would exert a head pressure of 100 psig on the pressure gauge at the bottom of the cylinder. The head pressure would be proportionally reduced as the liquid becomes less dense. For example, 231 ft of kerosene (which has 0.75 s.g.) would exert a head pressure of only 75 psig, because it is lighter than water.

34.3.4 Effect of Specific Gravity

Figure 34.9 shows a centrifugal pump that may pump water through valve B or naphtha through valve A.

Question: When water is pumped, the pump's discharge pressure is 110.4 psig. The pump's suction pressure is 10.4 psig. When naphtha is pumped, what is the pump's discharge pressure?

Answer: The head pressure developed by the pump is

$$\Delta P = (110.4 - 10.4) \times \frac{0.60}{1.00} = 60 \text{ psi}$$

and the pump discharge pressure = $10.4 + 60 = 70.4$ psig.

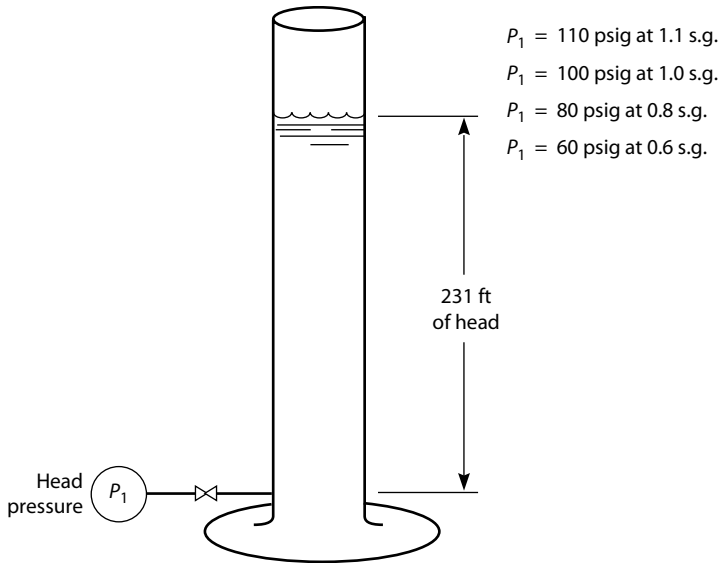


FIGURE 34.8 Head pressure compared to feet of head.

Next question: The pump is driven by an ordinary alternating-current electric motor. It is running at 3600 rpm. When we switch from pumping water to naphtha, what happens to the speed of the pump?

Answer: Nothing. An AC motor is a fixed-speed machine. It will continue to spin at 3600 rpm.

Next question: The pump is pulling 100 amps of electrical power. The flow rate is 100 GPM. If we now close valve B and open valve A, what happens to the amp load on the motor driver?

Answer: The power demand will go down to 60 amp. Amp(ere)s are a form of electrical work. The units of work are foot-pounds. The feet of head developed by the pump is not affected by the specific gravity of the liquid. But the weight of liquid pumped is proportional to the specific gravity. If the specific gravity drops by 40 percent, and the liquid volume (GPM) stays constant, then the pounds lifted by the pump drops by 40 percent and so does the electrical work.

Last question: Water is being pumped to a tank on a hill at 100 GPM. If we now switch to pumping a lighter fluid at the same rate, can we pump the lighter fluid to a higher elevation, only to a lower elevation, or the same elevation?

Answer: The same elevation. Centrifugal pumps develop the same feet of head at a given volumetric flow rate regardless of the specific gravity of the liquid pumped. This means the ability of the pump to push liquid uphill is the same even if the density of the liquid changes.

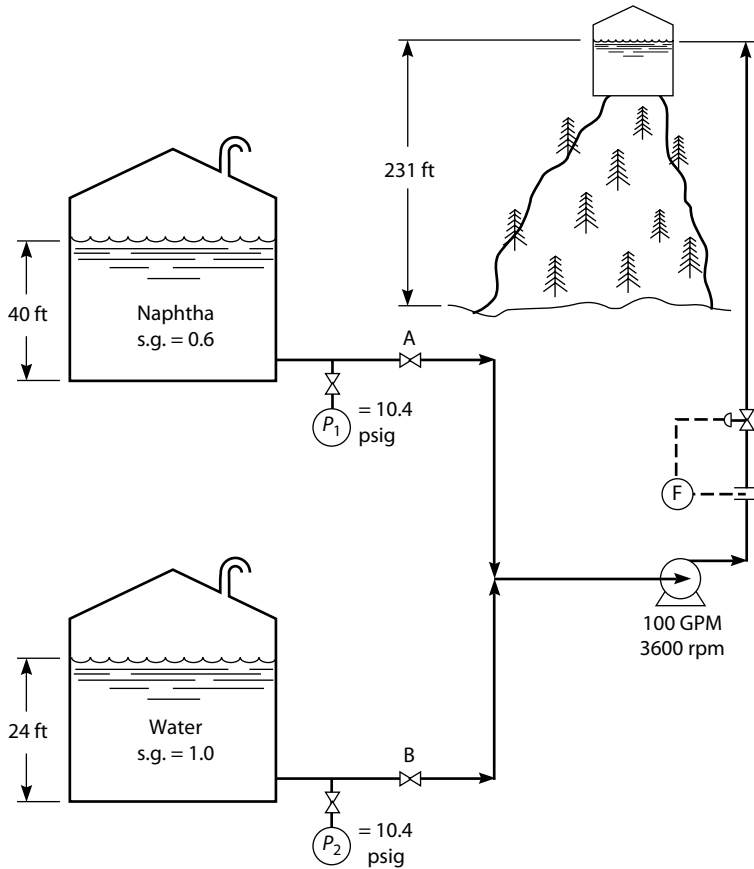


FIGURE 34.9 Effect of specific gravity on pump performance.

34.3.5 Pump Curve

The feet of head developed by a pump is affected by the volume of liquid pumped. Figure 34.10 is a typical pump curve. As the flow of liquid from a centrifugal pump increases, the feet of head developed by the pump goes down, as does the pump discharge pressure.

A pump curve has two general areas. These are the flat portion of the curve and the steep portion of the curve. We normally design and operate a pump to run toward the far end of the flat portion of its curve. A centrifugal pump operating on the flat portion of its curve loses only a small portion of its discharge pressure when flow is increased. This is a desirable operating characteristic.

A centrifugal pump operating on the steep portion of its curve loses a large portion of its discharge pressure when flow is increased.

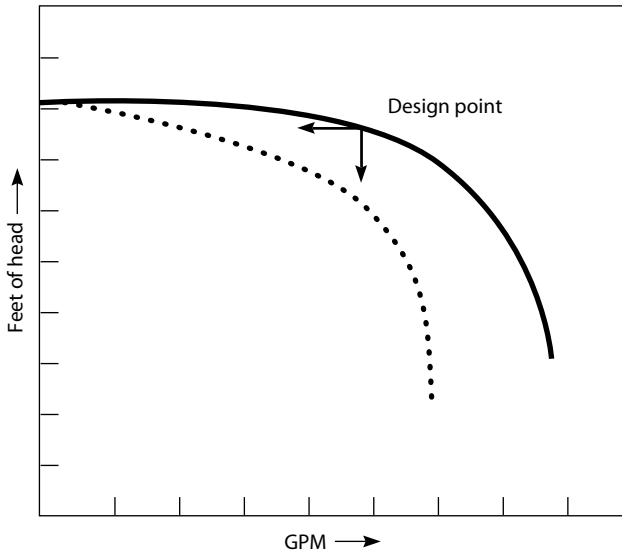


FIGURE 34.10 Centrifugal pump curve.

Pumps operating quite far out on their curve will have an erratic discharge pressure, as the flow through the pump varies. For one such pump, I have seen the discharge pressure drop from 210 to 90 psig as the flow increased by 40 percent. The flow through this pump was regulated by a downstream level-control valve. The discharge pressure varied so erratically that the operators thought the pump was cavitating.

34.3.5.1 Deviation from Pump Curve

Pumps always perform on their performance curve. It is just that they do not always perform on the manufacturer's performance curve. The dotted line in Fig. 34.10 compares an actual performance curve to the manufacturer's performance curve. Why the deviation?

- The manufacturer's curve was generated using water. We are pumping some other fluid with a much higher viscosity.
- The clearance between the impeller and the impeller wear ring (Fig. 34.6) may have increased as a result of wear.
- The impeller itself may be worn, or the vane tips at the edge of the impeller might be improperly machined.
- Perhaps the most common reason for the deviation is that the clearance between the impeller sides and the vertical sides of the pump case have increased, due to erosion of the interior of the pump case. This requires that the inside of the pump

case be built up with the weld metal, and then machined back to the original manufacturer's tolerances. This procedure is a lot of work and is often neglected by the maintenance department, who do not consider this "work" to be part of their job description.

- If the pump is driven by a steam turbine, the pump speed could be lower than in the design.
- Available NPSH may be marginal (see Chap. 36).

34.3.6 Driver Horsepower

Normally, increasing the flow from a centrifugal pump increases the amperage load on the motor driver, as shown in Fig. 34.11. Driver horsepower is proportional to GPM times feet of head. As shown in Fig. 34.10, as the flow increases, the feet of head developed by the pump decreases. On the flat part of the pump curve, the flow increases rapidly, while the head slips down slowly. Hence the product of GPM times feet of head increase.

On the steep part of the pump curve, the flow increases slowly, while the head drops off rapidly. Hence, the product of GPM times feet of head remains the same, or goes down.

The response of the amp load on the motor driver is an indication of pump performance. A pump operating on the proper, flat portion of its curve will pull more motor amps as flow increases. A pump operating on the poor, steep portion of its curve will not pull more amps.

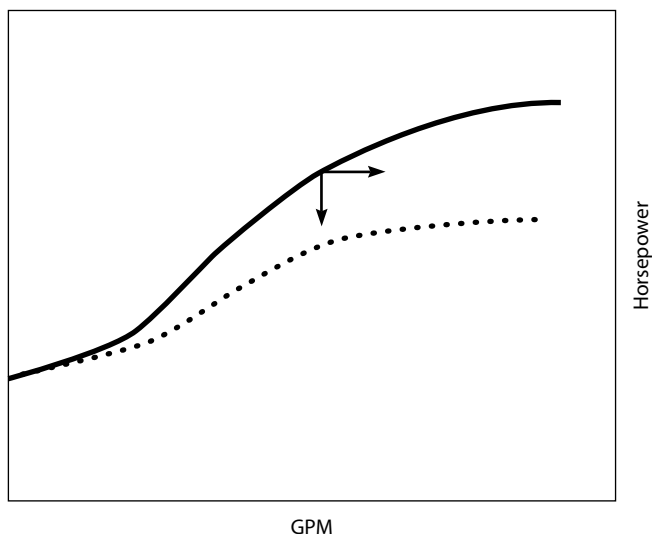


FIGURE 34.11 Horsepower requirement for a centrifugal pump.

If the manufacturer's horsepower/flow curve indicates that more horsepower is required as flow is increased, but the motor amp load is not increasing, then something is wrong with the pump.

The amp meter for a pump is best located right on the start-stop box next to the motor. Unfortunately, more often, the only amp meter is located on the electrical breaker box, which is at a remote location.

34.4 Pump Impeller

34.4.1 Impeller Diameter

The pump impeller resembles a hollow wheel with radial vanes. The diameter of this wheel can be trimmed down on a lathe. If an impeller with a 10-inch diameter is trimmed down to 9 inch, it will have a new performance curve below its former performance curve. Both the flow and head of the pump will be reduced. Also, the amp load on the motor driver will decrease. As a rough rule of thumb, reducing a pump's impeller diameter by 10 percent will reduce the amp load by 25 percent. Not only will this save electricity, but the motor will run much cooler. Unloading the motor by 25 percent may increase the life of the motor windings by 10 years.

When the control valve downstream of a pump is operating in a mostly closed position, the pump is a good candidate to have its impeller trimmed. Sometimes, the pressure drop across a control valve is so huge (≥ 100 psi) that it makes a roaring sound. The energy represented by this wasteful ΔP is coming from the electricity supplied to the pump's motor. Hence, trimming the impeller also reduces wear because of erosion in downstream control valves.

We frequently increase the size of impellers to increase pump capacity and discharge pressure. However, many chemical plants break the law when they do so. State *boiler codes* require that vessels and heat exchangers downstream of pumps be rated for the

- Maximum pump discharge pressure
- At the maximum specific gravity that can reasonably be expected
- Unless the equipment is protected by a relief valve

Increasing the size of an impeller always increases its maximum discharge pressure. Hence, increasing the impeller diameter may be unlawful, depending on the maximum allowable working pressure (MAWP) of downstream heat exchangers. The MAWP is shown on the exchanger nameplate.

Increasing the size of the impeller by 10 percent will increase the amperage load on the motor driver by 30 percent. For many process

pumps, this would require a new motor and breaker to support the larger impeller.

Trimming the pump impeller on a lathe also requires the reshaping of the tips of the impeller vanes. Most often, this requirement is ignored, and the revamped pump operates below its expected curve. Rather than worry about these fine points, it is best to order new impellers directly from the manufacturer.

Installing a larger impeller may require an upgrade in the pump bearings and the mechanical seal. While it is good practice to purchase a pump with a seal and bearings rated for the maximum impeller size, some pumps can have bearings and seals rated only for their purchased impeller size. For process people, the only practical method to check that a pump's seal and bearings are suitable for the larger impeller is to check with the manufacturer. Drilling pressure balancing holes in the hub of the impeller helps to reduce the extra strain on the bearings of a larger impeller.

Sometimes installing a full-sized impeller causes the pump to vibrate. This happens because of a disturbance to the liquid as the vanes spin past the volute or diffuser (i.e., the case outlet opening). Exxon had, at one time, a standard of keeping impeller sizes at not more than 92 percent of the full impeller diameter. This seems a conservative practice to me. I have usually kept impeller sizes in my revamp designs 0.25 to 0.5 inches below maximum. I don't recall any of my clients complaining that such pumps vibrated as a result of the enlarged impeller. On the other hand, for a new installation, now that I've learned of the Exxon standard, I plan to follow it in my process design work.

34.4.2 Impeller Rotation

Motor-driven pumps often are found running backward. If the motor is wired up backward (this is called reversing the polarity of the leads), the pump will run with the impeller spinning backward. This will reduce the pump's discharge pressure, sometimes by a little (10 percent) and sometimes by a lot (90 percent). It depends on the design of the impeller. You cannot see the direction of rotation of a pump, but if you touch a pencil to the spinning shaft, you can feel the direction of rotation. The correct direction of rotation is indicated by an arrow stamped on the top of the pump case. There are right- and left-handed pumps. So two otherwise identical pumps may have opposite directions of rotation.

A pump impeller will spin backward because the motor wiring is connected incorrectly. I'm referring to a three-phase motor. For a single-phase motor, the wiring connections may be reversed without affecting the direction of rotation.

In North America, we use 60 Hertz electric power and motors operate at 1800 rpm (single phase) or 3600 rpm (three phase). In Europe

and Asia, 50 Hertz electric power is the standard; single-phase motors spin at 1500 rpm and three-phase motors at 3000 rpm.

I was working on a project to relocate a U.S. process unit to Cyprus. The project met a merciful death when I explained that a 100-horsepower American pump (60 Hertz) would only produce 58 percent of pumping power when connected to a European (50 Hertz) motor. That is:

- $(100 \text{ hp}) (50/60)^3 = 58 \text{ hp}$

According to the affinity or fan laws, work or power varies with speed, cubed.

It is possible to rewire a three-phase motor into a single-phase motor, and thus reduce its speed from 3600 to 1800 rpm. Then its power consumption would drop by a factor of 8. I've done this once at an East Coast refinery, which had a hugely oversized sulfur plant incinerator air blower. For most projects, a more typical option would be to retrofit the motor for variable-speed operation, using frequency control. However, not all motors are amenable to frequency control, which is a relatively expensive option. Feedback from my clients that have retrofitted pump drivers with variable-speed AC motors is often unfavorable.

34.5 Effect of Temperature on Pump Capacity

If we wish to increase the capacity of a centrifugal pump, should we make the liquid hotter or colder? Let's make a few assumptions:

- Cooling the liquid will not increase its viscosity above 30 or 40 centistokes.
- We are pumping hydrocarbons. Cooling many hydrocarbons by 100°F increases their density by 5 percent.
- The pump's motor is somewhat oversized.

On this basis, cooling the liquid by 100°F will raise the pump's discharge pressure by 5 percent. If the pump is developing 1000 psig of differential pressure, cooling the liquid by 200°F will increase its discharge pressure to 1100 psig. If the discharge control valve is now opened, the pump's discharge pressure will drop back to 1000 psig as the flow increases. The discharge pressure drops because the pump develops less feet of head at a higher flow rate.

To summarize, the specific gravity of the liquid has increased by 10 percent. The feet of head of liquid has decreased by 10 percent. Since:

$$\Delta P \approx (\text{specific gravity}) \times (\text{feet of head})$$

ΔP remains constant. But the lower required feet of head permits the pump to operate further out on its performance curve. This produces

more volumetric (GPM) flow. It also requires more work to pump the greater flow. So we have to have sufficient excess amperage on the motor driver to accommodate this maneuver. But that is the price we pay for expanding the pump's capacity without increasing the impeller diameter. Of course, the larger-diameter impeller would also require more motor amperage.

CHAPTER 35

Centrifugal Pumps: Driver Limits

Electric Motors and Steam Turbines

There are three types of limits on centrifugal pumps:

- *Impeller limit.* This was the subject of Chap. 34. Sometimes this is called a *pump limit*.
- *Driver limit.* This means the electric motor is tripping off on high amperage or the turbine is running below its set speed.
- *Suction pressure limit.* Usually called lack of *net positive suction head* (NPSH), this is the subject of Chap. 36, which follows.

35.1 Electric Motors

Pumps are driven by either fixed-speed or variable-speed motors. Variable-speed motors are becoming increasingly common. Flow can be controlled by varying the pump speed with the motor and thus eliminating parasitic energy losses across a flow control valve. However, because of the extra electronic components needed for a variable-speed AC motor, 95 percent of the motors we work with are fixed-speed alternating-current motors. I will only discuss fixed-speed, alternating-current (AC) motors in this chapter.

Direct-current (DC) motors are sometimes used in process plants. It is easy to manipulate the speed of a DC motor. But direct current is almost never available on a process unit. Our motors at home are mainly alternating-current motors operating at a fixed speed.

A process motor in an American, Canadian, or Latin American plant is likely to run at either 1750 to 1800 rpm or 3500 to 3600 rpm. In other locations, a motor will likely run at 1500 or 3000 rpm.

In the United States, we use alternating current at 60 Hz cycles per second. In Europe, they use alternating current at 50 Hz cycles per second. As the alternating current cycles 20 percent slower in Europe, the motor spins 20 percent slower.

At home, our motors are single phase. At the plant, the motors are usually three phase. There is an important difference between a single- and a three-phase motor. The three-phase motor can also be used as an electric power generator. The single-phase motor we use at home, without modification, cannot be used to generate electric power. The difference is important in that I once used this fact to avoid tripping off a critical process pump, as related in the following story.

35.1.1 Helper Turbine

Figure 35.1 is a sketch of a centrifugal pump driven by a three-phase motor with a turbine helper. This particular pump was charging a light gas oil stream to a high-pressure hydrocracker. The pump was operating quite close to its design conditions of

- $\Delta P = 2000$ psig
- Flow = 17,000 bbl/day
- Specific gravity = 0.80
- Temperature = 400°F
- Speed = 3600 rpm

The operators were not operating the turbine. The turbine was spinning, because it was coupled to the pump, but there was no

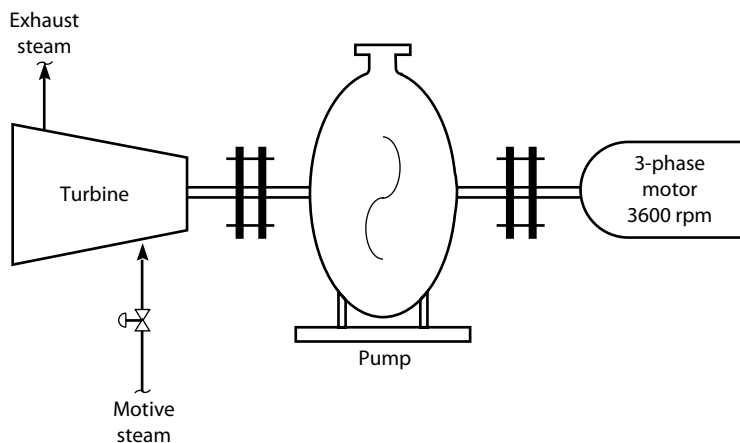


FIGURE 35.1 A pump driven by a motor with a turbine helper.

motive steam to the turbine. The operators reported that the turbine was not needed, as the motor was pulling only 90 percent of its maximum amperage load. The question is, dear reader, whether the pump will run faster if the motive-steam flow is opened to the turbine. And the answer is no. While the turbine will produce shaft work, and will help the motor spin the pump, the pump speed will remain unchanged. The extra shaft work from the turbine will just reduce the shaft work required from the motor shown in Fig. 35.1. The amp(era)ge load on the motor will just be reduced.

But suppose the amp load on the motor drops to zero, by gradually increasing the steam to the turbine. Then let's suppose we increase the motive steam to the turbine by another notch. Will the pump, the motor, and the turbine (which are all coupled together and, hence, must run at the same speed) now run faster? The answer is no. But what will happen to the increment of shaft work generated by the turbine?

This extra shaft work will be converted to electrical power. That is, the three-phase motor will turn into an electrical power generator. It will begin to export electricity into the grid. A single-phase motor does not have this sort of capability.

Another way of expressing this idea is that the three-phase motor acts as a brake on the turbine. The motor can run at only 3600 rpm as long as it is connected to the electrical grid. The only way we can get more flow from this pump without reducing the pump's ΔP is to raise the specific gravity of the liquid being pumped. If the liquid is cooled from 400 to 200°F, its specific gravity will increase by about 10 percent. This will allow us to get the same ΔP with about 10 percent less feet of head. This will allow the pump to run further out on its performance curve. The pump will now pump more liquid, and the flow of light gas oil to the hydrocracker could be—and was—increased from 17,000 to 23,000 bbl/day.

Question: How does this affect the amp load on the motor driver? Well, the feet of head has gone down by about 10 percent, and the specific gravity has gone up by about 10 percent. Therefore, the horsepower or work per barrel of liquid pumped remains constant. But the number of barrels has increased by 6000 bbl/day, or 35 percent. Then the amp load on the motor driver would also increase by 35 percent. If I had not remembered this prior to cooling the hydrocracker feed, the motor would have tripped off. As it was, I had the steam to the turbine increased so as to decrease the motor amps to 60 percent of maximum before cooling the gas oil. Omission of this detail would have had embarrassing and possibly negative contractual consequences.

35.1.2 Selecting Motor Size

We noted in Chap. 34 that increasing the impeller diameter might require a larger motor for an existing pump. Suppose, then, when a new pump is purchased, we install a 100-hp motor sized for the

maximum 10-inch impeller, which the pump could accommodate. Actually, only an 8.5-inch impeller would be used in the pump initially. This 8.5-inch impeller requires only a 60-hp motor. If we install a 100-hp motor, what percent of the electrical power of the oversized motor is wasted?

Essentially, none. The amp load on the motor driver is determined by the work done by the pump, not by the size of the motor. The amp load required by the motor is proportional to

$$\frac{(\text{Pump head}) \times (\text{specific gravity}) \times (\text{liquid flow})}{(\text{Motor efficiency})}$$

There will be a small loss of 3 percent in motor efficiency by using an oversized motor. As process operators and engineers, we can ignore this effect. It is good engineering practice to purchase new motors for the maximum-size impeller that can be installed in a pump.

35.1.3 Motor Trip Point

Factors that increase the amp load on a pump's motor driver are

- Increasing the flow, provided the pump is running on the flat portion of its curve.
- Increasing the specific gravity of the liquid pumped.
- Increasing the impeller size. A 10 percent increase in impeller size may increase the motor amp load by 33 percent.
- Motor winding deterioration, which happens as the motor ages, depending on how hot the motor runs.
- Dirt buildup on the motor's cooling-air fan guard screen.

Most operators forget to clean this screen. The dirty screen then restricts air flow to the motor, which then runs hotter. A hotter motor will pull more amps. I recently reduced the amp load on a motor by 3 percent by cleaning a badly fouled screen.

The amp meter on the motor breaker indicates the amps being pulled by the motor. Next to this meter should be a tag or a penciled number indicating the full-limit amp (FLA) point for the motor. Above the FLA point, the motor should trip off. Operating the motor substantially above its FLA point will burn up the motor's windings.

This is not quite true. Whenever we start a pump, the starting torque required to get the pump spinning requires a surge of motor amps. To avoid tripping off the motor on high amps, there is a time delay built into the trip mechanism. This delay permits the amperage load to greatly exceed the FLA point for up to 15 to 30 seconds. This is too brief a period in which to overheat the motor.

If we cannot open the discharge control valve of a centrifugal pump 100 percent before the FLA point is reached, then we say that the pump is *driver-limited*. This is a frustrating problem for plant operators, and clearly reflects poor design practice in undersizing the motor driver horsepower.

35.1.4 Internal Motor Thermal Protection

I have made the statement earlier in this chapter that alternating-current motors operate at a constant speed. I wrote this because I've read it in a book. Yet, I know from personal experience that this is wrong. I know that my wood-turning lathe will slow down if I press too hard on the spinning wood with my cutting chisel. The lathe is run by a 1750-rpm, 60-Hz, 120-V, 3/4-hp AC motor. But I also know that if I run the motor at a slower speed for a minute or two, the motor will get hot and trip off on its internal thermal trip switch. I will then have to wait 10 minutes for the motor to cool off before I hear the click of the trip resetting.

The motor has gotten hot because an alternating-current motor efficiency goes down a lot when it runs below its rated speed. That is, running an alternating-current motor slow increases the motor amperage requirement. The extra amperage goes to heat and not to increased motor work output.

At the plant, our motors are protected differently than at home. At home, our motors share a circuit with other power consumers. A circuit breaker may be rated for 20 amps, but a single motor on that circuit may be rated for 5 amps. The circuit breaker setting of 20 amps is too large to protect the motor from overheating. At the plant, each motor has its own dedicated circuit breaker, set individually to protect a particular motor from overheating (i.e., the full-limit amperage point). If the load on the motor increases too much because the specific gravity of the fluid pump has increased, the motor may indeed momentarily slow down. But due to the lower efficiency at the slower speed, the amperage load will increase rapidly as the alternating-current motor slows. The FLA point on the circuit breaker in the breaker or switchgear house for that motor will be exceeded and the motor will trip off.

To summarize, at home, motors can run slower for a minute or two before they trip off on high winding temperature. At work, motors trip off at the breaker after a few seconds of running slow due to the high amperage load exceeding the breaker full-limit amperage setting of the breaker.

35.1.5 Motor Bearings

If you think a motor is running rough because of bad bearings, have it uncoupled from the pump. Run the motor alone, and see if the vibrations persist. Motor bearings commonly fail because of a lack of lubrication.

Most motors are lubricated with a heavy grease injected through a grease fitting delivered via a grease gun. Overgreasing can burn up the motor bearings. You need to follow the manufacturer's recommendations. A typical (but not general) program for motor bearing lubrication is three or four squirts from a grease gun every six months. There is a device now available on the market called a "bearing buddy," which delivers a consistent grease supply pressure over an extended period of time. Motor bearing failures on one process facility disappeared when these bearing buddies were installed.

35.2 Steam Turbines

35.2.1 Turbine Drives

A turbine-driven pump is said to be driver-limited when the governor speed-control valve is wide open. This speed-control valve is usually called the "Woodward governor." It is not that easy to see if the governor is really wide open. A few simple methods to make this determination are

- Increase the pump's flow. If the turbine slows, the governor is wide open.
- Increase the set-point speed. If the turbine speed fails to come up, the governor was already wide open.
- Throttle on the steam supply to the governor. If the governor is already wide open, the turbine will slow down.

Even if the governor-steam inlet control valve is 100 percent open, it may still be possible to increase the supply of motive steam into the turbine. The position of the governor when it is wide open can often be reset to admit more steam flow. I once increased the flow through a crude tower-bottom pump by pushing hard against the base of the governor, and forced it to actually open to its maximum position. The pump speed came up 300 rpm, and the flow increased by about 15 percent.

Another possibility is to open the speed or hand valves, as described in Chap. 24, "Steam Turbines." This will introduce more steam into the turbine case and slightly lower the pressure in the steam chest. Even though the governor-steam speed-control valve is wide open, the reduction in downstream pressure in the steam chest will increase the steam flow into the turbine.

Of course, increasing the motive-steam pressure will greatly increase the horsepower available to the turbine. First, more work can be extracted from each pound of the motive steam; and second, increasing the steam pressure will increase the pounds of steam flowing through the governor.

Reducing the turbine exhaust pressure will not significantly increase the steam flow to the turbine. However, reducing the turbine exhaust-steam pressure will increase the amount of work that may be extracted from each pound of steam. For example, 150-psig, 400°F steam is being used to drive a turbine. The exhaust steam is flowing into a 15-psig steam header. By venting the steam to the atmosphere, the amount of work that can be extracted from each pound of steam will increase by roughly 30 percent (see the Mollier diagram).

35.2.2 Increasing the Size of Steam Nozzles

Many steam turbines do not have full-ported steam nozzles (see Chap. 24). The existing steam nozzles may then be exchanged for larger nozzles. An increase of nozzle diameter of 10 percent would increase the turbine horsepower by 20 percent.

Finally, the steam turbine's buckets can foul with hardness deposits from the steam. This reduces the turbine efficiency and may prevent a pump from running at its rated speed. Injecting steam condensate into the steam supply can remove such deposits.

It is quite important not to operate a turbine-driven pump by manually throttling the steam flow to the turbine. Let's assume that the operators have set the turbine speed at 3500 rpm by adjusting the steam inlet gate valve upstream of a malfunctioning governor. Suddenly, the discharge flow-control valve cuts back, and the pump's flow decreases from 2000 to 1200 GPM. The pump speed will then increase, because fewer pounds of liquid are being pumped and less horsepower is required to spin the pump.

The pump speed rises to 3800 rpm. The trip speed has been set at 3750 rpm. The turbine's overspeed trip is unlatched, and the machine shuts down. The operator relatches the trip, but every time the flow is throttled back, the turbine overspeeds, and trips off.

Finally, the operator, in frustration, decides to wire up the trip so that it cannot unlatch. As an operator in a Gulf Coast refinery stated, "Norm, when the governor won't work on a turbine, it is necessary to wire up the trip." I thought he was joking, but he could not have been more serious.

Well, the operator did have a pump with an inoperable governor speed-control valve. He did wire the trip open. A few days later, the pump briefly lost suction, due to a slightly low level in the flash drum feeding the pump. The pump cavitated, and the flow was reduced. Reducing the flow to or from a centrifugal pump reduces the horsepower load on the driver. As the steam flow to the turbine driver was fixed, the turbine started to overspeed. The excessive speed damaged the pump's bearing. The resulting vibration fractured the seal flush oil line to the pump. The flush oil ignited on the 700°F pump case. A nearby worker was sprayed with the burning oil and killed! This is a true story.

35.3 Gears

On occasion, pumps are not directly connected to either a motor or a turbine. There is an intervening gear, which can increase the pump's speed by multiplying the driver's speed. The gear is another source of possible misalignment and vibration. I have always considered such reduction or speed increaser gears to represent poor design practice, and an unnecessary complication as they often require their own lubrication system. Most often the lube oil pump is driven indirectly by the main pump. Especially for a turbine-driven pump, this is a recipe for disaster.

It's true that the primary lube oil pump is typically backed up by a separate, spare lube oil pump driven by an external power source. However, only too often that backup pump, which is rarely used, fails to start up automatically due to lack of routine testing, and the gears are destroyed.¹

Reference

1. N. P. Lieberman, *Troubleshooting Process Plant Control* (Wiley, 2009).

CHAPTER 36

Centrifugal Pumps: Suction Pressure Limits

Cavitation and Net Positive Suction Head

The single most common operational problem in a process plant is loss of suction pressure to a centrifugal pump. If the suction pressure is too low, the discharge pressure and the discharge flow become erratically low. The suction pressure, while low, remains comparatively steady.

36.1 Cavitation and Net Positive Suction Head

The problem described above is called *cavitation*. A pump that is operating in a cavitation mode may also (but often does not) produce a sound similar to shaking a bucket full of nuts and bolts. A cavitating motor-driven pump always draws an erratically low-amperage flow. This is consistent with the erratically low flow rate.

(*Caution:* When a pump completely loses flow, it runs smooth, steady, and quiet. Its discharge pressure is stable. Its discharge flow is also stable and steady—a steady zero. This is not cavitation. The pump impeller is just spinning, to no particular purpose, in the empty impeller case.)

36.1.1 Causes of Cavitation

Let us assume liquid flows from an 8-inch line into the suction of a centrifugal pump. The liquid enters the pump's impeller through a circular opening, called the "eye of the impeller," in the center of the impeller. Let us assume that this eye has a diameter of 2 inches.

The velocity of the liquid increases by a factor of 16 (i.e., 8 inch \div 2 inch, squared). The kinetic energy of the liquid increases by a factor of 264 (i.e., 16 squared). But where does this large increase in kinetic energy come from? Answer—from the pressure, or feet of head, of the liquid itself.

The conversion of the pump's suction pressure to velocity in the eye of the impeller is called the *required net positive suction head* (NPSH). As the flow-control valve on the discharge of the pump shown in Fig. 36.1 is opened, the velocity of liquid in the eye of the impeller goes up. More of the pump's suction pressure, or feet of head, is converted to velocity, or kinetic energy. This means that the required NPSH of a pump increases as the volumetric flow through the pump increases.

The units of NPSH are feet of liquid head. The required NPSH of a pump is due primarily to the conversion of feet of head to velocity in the eye of the impeller.

The *available NPSH* to a pump has the following definition:

- Physical pressure pump at suction
minus
- Vapor pressure of liquid at pump suction

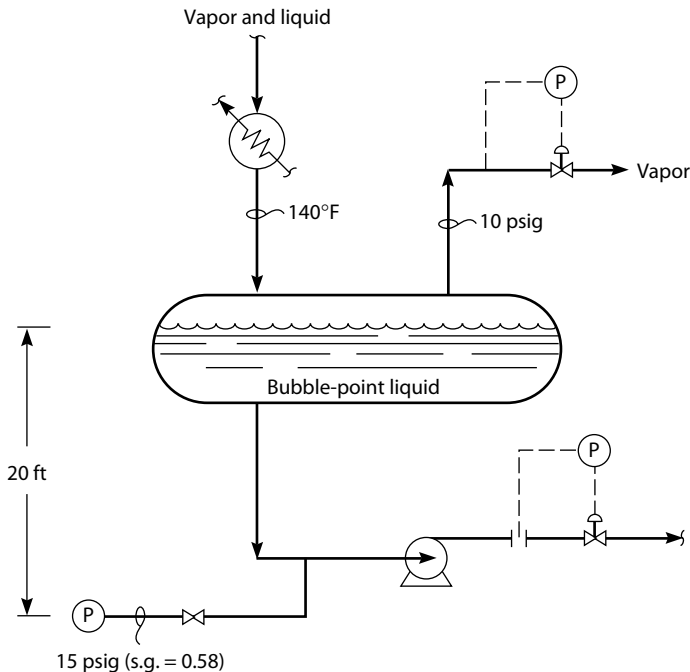


FIGURE 36.1 Available NPSH equals 20 ft.

To express the available NPSH in feet:

- Measure the pump suction pressure, and add the barometric pressure (14.67 psia at sea level).
- Look up, on a vapor pressure chart, the vapor pressure of the liquid pumped, at the measured pump suction temperature, in psia.
- Multiply the difference between the two above pressures by 2.31 ft (for water).
- Divide by the specific gravity at flowing conditions of the liquid (1.00 for water, 0.85 for diesel, 0.50 for propane) at the pump suction temperature.

Note that the pump suction pressure is the pressure downstream of the pump's suction screen, and not a pressure read from a control panel screen.

When the required NPSH of a pump equals the NPSH available to the pump, the pump will cavitate or slip.

36.1.2 Cavitation Illustrated

Let us see what *cavitation* means in reference to the pump shown in Fig. 36.1. The liquid shown in the vessel is presumed to be in equilibrium with the vapor leaving the drum. This means that the liquid is at its bubble-point pressure and the vapor is at its dew-point temperature.

The vapor pressure of the liquid is then 24.7 psia (10 psig plus 14.7 psia of atmospheric pressure). The physical pressure at the suction of the pump is measured at 29.7 psia (15 psig plus 14.7 psia of atmospheric pressure).

Then the physical pressure at the suction of the pump of 29.7 psia minus the vapor pressure of the liquid at the suction of the pump of 24.7 psia is 5 psia. This will be the NPSH available to the pump after we convert from psia to feet of liquid head.

The specific gravity of the liquid being pumped, as shown in Fig. 36.1, is 0.58 s.g. To convert from psia to feet of liquid head, we have

$$5 \text{ psia} \times \frac{2.31}{0.58} = 20 \text{ ft of head}$$

The 20 ft matches the level of liquid in the drum above the suction line of the pump, shown in Fig. 36.1, and equals the available NPSH to the pump.

The required NPSH of the pump may be read from Fig. 36.2 (regardless of the s.g. of the liquid being pumped). It shows that at 250 GPM, the required NPSH of 20 ft will equal the available NPSH of 20 ft. Therefore, at a flow rate of 250 GPM, the pump will cavitate. This calculation has neglected frictional losses in the suction line and nozzle, which should be subtracted from the available NPSH.

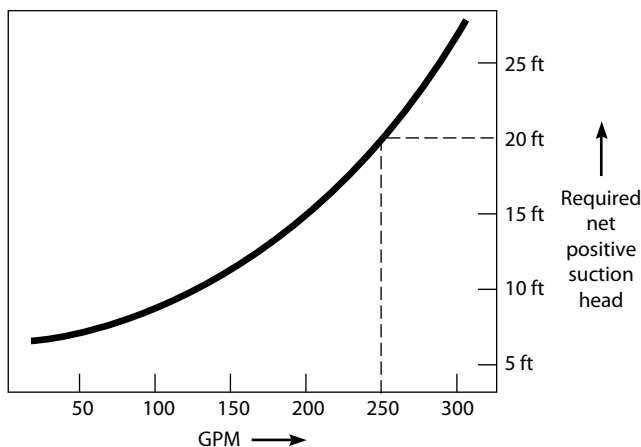


FIGURE 36.2 Required NPSH increases with flow.

Let's now assume that we wish to pump 300 GPM, not 250 GPM. If we open the flow-control valve shown in Fig. 36.1, the flow will momentarily increase. But within a few seconds, the flow will become erratically low as the pump begins to cavitate. The problem is that, according to Fig. 36.2, we require an additional 6 ft of NPSH to increase the flow from 250 to 300 GPM.

One way of getting this extra suction pressure, or NPSH, is to raise the liquid level in the drum. For a liquid of 0.58 s.g., with every 4 ft we raise the level in the drum, the suction pressure will increase by 1 psi, and the available NPSH will increase by 4 ft. But, unfortunately, the drum shown in Fig. 36.1 is already almost full.

Suppose we increase the pressure by partly closing the back-pressure control valve. This will quickly increase the pressure in the drum from 10 to 15 psig. The pressure at the suction of the pump will also increase, from 15 to 20 psig. However, will this provide more NPSH to the pump?

Answer—no! Unfortunately, it is not only the pressure of the liquid at the pump that changes. The composition of the liquid will also be altered. As the pressure in the drum increases, additional lighter components dissolve in the liquid. The composition of the liquid then becomes lighter. The vapor pressure of the liquid will also increase by 5 psi. This must happen because the liquid in the drum, which is in equilibrium with the vapor, is at its bubble-point pressure.

Again, the available NPSH is the physical pressure at the suction of the pump minus the vapor pressure of the liquid at the suction of the pump. If both pressures increase by 5 psi, the net gain in NPSH is zero.

Let us try again. Suppose we decrease the temperature of the drum, shown in Fig. 36.1, from 140 to 110°F. This will also cool the liquid flowing into the suction of the pump by 30°F. Will this colder liquid then provide more NPSH to the pump by suppressing the flashing of the liquid flowing into the pump's impeller?

Answer—no! Unfortunately, it is not only the temperature of the liquid that changes. The composition of the liquid has also been altered. As the temperature in the drum decreases, additional lighter components dissolve in the liquid. The composition of the liquid becomes lighter. The liquid will now boil, not at 140°F, but at 110°F. This must happen because the liquid in the drum, which is in equilibrium with the vapor, is at its bubble-point temperature, which is now a cooler 110°F.

Let's try one more time. Suppose we tear off the insulation on the suction line and on the pump case and then spray cool water on the bare line. The temperature of the liquid in the drum, which is at 140°F, cools as it flows into the pump. By the time the liquid reaches the eye of impeller, it has cooled to 135°F. Will this slightly colder liquid provide more NPSH to the pump?

Answer—yes! But why? Well, the liquid is cooled by 5°F after it leaves the drum. The cooled liquid is not in equilibrium with the vapor in the drum. It has been *subcooled* by 5°F. This means that the bubble-point liquid has been cooled without altering its composition. The vapor pressure of the liquid has been reduced. As can be seen in Fig. 36.3, subcooling this particular liquid by 5°F reduces its vapor pressure by about 2 psi. As the specific gravity of the liquid is 0.58, this is equivalent to an increase in the NPSH by 8 ft. Once again, our objective is to increase the flow from 250 to 300 GPM. Figure 36.2 tells us that the required NPSH increases from 20 to 26 ft. However, when

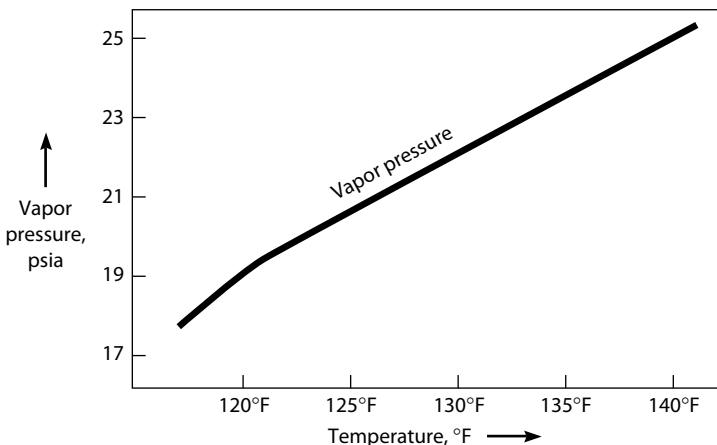


FIGURE 36.3 Subcooling increases available NPSH.

we subcool the liquid by 5°F, the available NPSH increases from 20 to 28 ft. As the available NPSH now exceeds the required NPSH by 2 ft, the flow can be increased without risk of pump cavitation.

36.1.3 Starting NPSH Requirement

The required NPSH read from the manufacturer's pump curve is called the *running NPSH*. However, when a pump is put on line, there is an additional type of NPSH requirement. This is called the *starting NPSH*. The initial velocity of the liquid in the suction of the pump is zero. After the pump is up and running, the velocity of the liquid in the suction of the pump might be 6 ft/s. This means that the liquid in the suction line has to be accelerated, which requires energy. This energy does not come from the pump; it must come from the liquid in the suction line itself. The only source of energy the liquid has is its pressure. This means that the pressure of the liquid is converted into kinetic energy. The kinetic energy accelerates the liquid from 0 to 6 ft/s.

This results in a temporary loss of pressure at the suction of the pump. This temporary loss of pressure is called the *starting NPSH requirement*. The more quickly the operator opens the discharge valve of a pump, the more rapidly the liquid accelerates in the suction line. This increases the starting NPSH required.

The longer the suction line and the larger the diameter of the line, the more mass has to be accelerated. This also increases the starting NPSH required. If the sum of the frictional loss in the suction line plus the running NPSH plus the starting NPSH equals the available NPSH, then the pump will cavitate on start-up.

The experienced design engineer always allows for the starting NPSH requirement when determining the elevation of a vessel. It is hard to say whether to allow an extra 2 ft or an extra 10 ft. It depends on the size of the suction line, and how careful the operators are likely to be when starting the pump. But if the designer forgets this factor, then the plant operators are sure to notice the omission when the pump slips during start-up (See sec. 34.2 for detailed calculations for starting NPSH).

36.1.4 A Hunting Story

Gentle reader, thank you for reading so much of our book. Liz and I know you must be tired of all this technology. So let's change the subject. Let's talk about hunting.

How many of you are hunters? How many of you enjoy killing the innocent creatures of the forest? Actually, I used to go hunting once a year in Casper, Wyoming. At that time, I was working for Amoco Oil. So, once a year at the start of the antelope season, I found an excuse to visit Amoco's refinery in Casper.

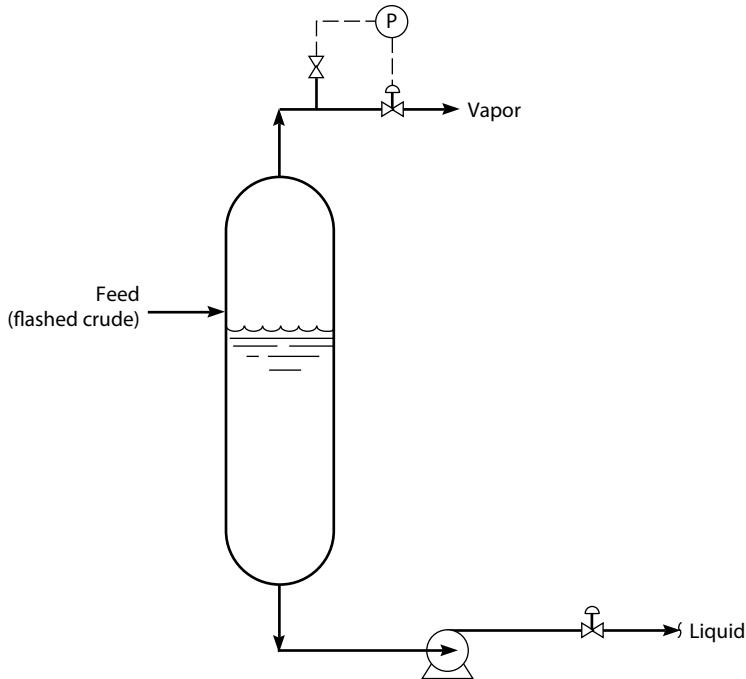


FIGURE 36.4 Overcoming a starting NPSH limit.

I remember my last visit, in 1980. My excuse to visit the refinery that year was to start up the pump shown in Fig. 36.4. This was a new, flashed crude-oil pump. It had never been run before. The refinery operators reported that the pump always cavitated on start-up. They had raised the liquid level in the drum to within a few inches of the inlet nozzle. A higher level would cause entrainment of the black crude oil into the vapor outlet. The operators had then started the motor driving the pump. Next they very slowly opened the discharge valve.

Regardless of their efforts, the pump cavitated so badly that it could not be put in service. Some pumps are like that. The design engineer just never allowed any extra liquid elevation for the starting NPSH. But what could I do?

You see, guys, here is the problem. I cannot go hunting until I get the pump going. And I do not have much time. My friends are waiting for me at the plant gate. They're waiting in those great big high-wheel pickup trucks designed for off-road use. They're blowing their horns and yelling for me to hurry up. It is a macho-type thing!

For my part, I cannot see the rush. I would just as soon stop at the gates of the city of Casper, Wyoming, and shoot the first antelope we came across. But not my friends. They want to go deep out into the back country to search for the "ghost antelope." And they are in a

big hurry to get started. What, then, can I do to get the flashed crude pump up and running in just a few minutes?

36.1.5 Temporary Increase in NPSH

This pump is presumed to run fine once it is running. The available NPSH is such that it exceeds the running NPSH. So how can I provide a temporary increase in the available NPSH, to satisfy the temporary starting NPSH requirement?

Answer—suddenly increase the pressure in the drum by partly closing the back-pressure control valve shown in Fig. 36.4. This will instantly increase the pressure at the suction of the pump. It is true, as we said before, that raising the pressure in a drum does not increase the available NPSH, assuming that the vapor and liquid are at equilibrium. The idea of equilibrium assumes that the vapor is at its dew point and the liquid is at its bubble point.

As soon as the drum pressure is raised, the vapor composition in the drum is altered. The vapor composition becomes lighter. The vapor, though, is still at its dew point.

As soon as the drum pressure is raised, the liquid composition in the drum is altered. The liquid composition at the vapor-liquid interface becomes lighter. The liquid formed at the vapor-liquid interface is still at its bubble point.

When this lighter liquid works its way down to the suction of the pump, the beneficial effect achieved by raising the pressure in the drum is gone. The available steady-state NPSH will be exactly what it was before the drum pressure was raised. But this will take *time*. If the residence time of the liquid in the drum is 10 min, then it will take 10 min for the lighter liquid to reach the suction of the pump.

During this 10-min interval, the liquid flowing into the pump is the older, heavier composition. If I raise the pressure in the drum suddenly by 5 psi, this instantly supplies about 20 ft of additional available NPSH—but only for a period of less than 10 min. During this period, I can crack open the pump discharge valve, push the motor START button, and then slowly accelerate the liquid in the suction line by slowly opening the discharge valve. If I can do all this before the lighter liquid—formed at the vapor-liquid interface in the drum—reaches the eye of the pump's impeller, then I can start up the pump. Thus, even though we lacked the extra available NPSH to satisfy the pump's starting NPSH requirement, I could still go antelope hunting with my friends.

My pal, “one-shot” Bob Boening, did manage to shoot an antelope with a single round. As we approached the dying creature, it staggered to its feet and ran off. Bob and I then spent the rest of the day crawling and clambering through the ravines and rocky hills south of Casper, searching for what did prove to be a “ghost antelope.”

36.1.6 Why Some Pumps Cavitate

Pumps cavitate for three reasons:

- They lack sufficient available NPSH to satisfy the conversion of pressure to velocity in the eye of the impeller (running NPSH).
- They lack sufficient available NPSH to satisfy the conversion of pressure to acceleration in the suction line as the pump is started (starting NPSH).
- They lack sufficient available NPSH to overcome the frictional losses in the suction piping and the drain or draw nozzle.

It is positively my experience that the most common reason for pumps' cavitation is partial plugging of draw nozzles. This problem is illustrated in Fig. 36.5. This is the side draw-off from a fractionator. Slowly opening the pump's discharge control valve increases flow up to a point. Beyond this point, the pump's discharge pressure and discharge flow become erratically low. It is obvious, then, that the pump is cavitating.

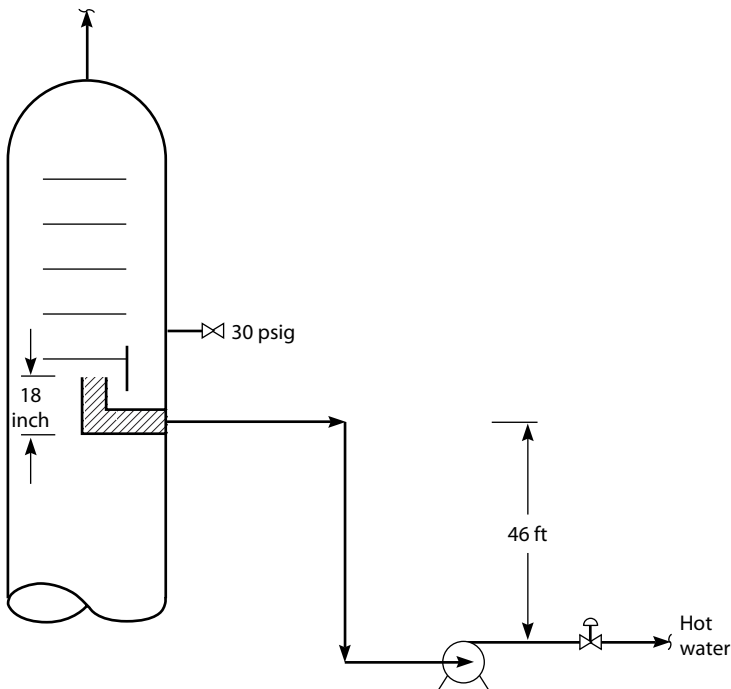


FIGURE 36.5 Partly plugged draw-off nozzle.

The fluid being pumped is hot water. At the desired flow rate of 110 GPM, the manufacturer's pump curve shows that the pump requires 14 ft of NPSH. The elevation difference between the draw-off nozzle and the suction of the pump is shown on Fig. 36.5 as 46 ft. We really ought to have plenty of running NPSH. But apparently, we do not.

If I reduce the flow of water by just 10 percent down to 100 GPM, the cavitation stops. I now put a pressure gauge on the suction of the pump. Assuming that the suction line is full of 46 ft of water, what suction pressure would I expect? Answer:

$$\frac{46 \text{ ft}}{2.3 \text{ ft/psi}} + 30 \text{ psig} = 50 \text{ psig}$$

The 2.3 ft/psi factor assumes that the specific gravity of water is 1.00.

But the observed pressure is not 50 psig. It is only 47 psig. I am missing 3 psig or 7 ft of liquid:

$$(50 \text{ psig} - 47 \text{ psig}) \times 2.3 \text{ ft/psi} = 7 \text{ ft}$$

The most likely explanation for this *head loss* of 7 ft is frictional loss in the suction line. This reduces the available NPSH from 46 to 39 ft. But this is still a lot more available NPSH than the 14 ft of required NPSH needed to pump 110 GPM.

If I now open the discharge flow-control valve sufficient to increase the flow from 100 to 110 GPM, or by 10 percent, this would increase the frictional loss in the suction piping by about 21 percent, or about 0.5 psi; ΔP varies with $(\text{flow})^2$.

But this is not what I observe. The suction pressure in Fig. 36.5 slips from 47 to 36 psig. At which point the pump begins to cavitate. What is happening? How could just a 10 percent increase in flow cause such a large increase in the suction line delta P ? What has happened to the lost 11 psi (i.e., 47 psig minus 36 psig) of suction pressure?

The boiling-point pressure of the water is equal to 30 psig (the pressure in the tower shown in Fig. 36.5); that is, we can assume that the water draw-off is at its bubble-point pressure. At 36-psig pump suction pressure, the available NPSH is then

$$(36 \text{ psig} - 30 \text{ psig}) \times 2.3 = 14 \text{ ft}$$

This matches the required NPSH at a flow of 110 GPM, so the pump cavitates. But it still seems as if I am missing at least half of the 46 ft of liquid head to the pump. Where is it?

Well, dear reader, it no longer exists. Figure 36.6 illustrates the true situation. Let's say we are pumping 110 GPM from the pump discharge. But only 109 GPM can drain through the draw-off nozzle. We would then slowly lower the water level in the suction line. The water level would creep down, as would the pump's suction pressure. When the water level in the suction line dropped to 14 ft, the pump

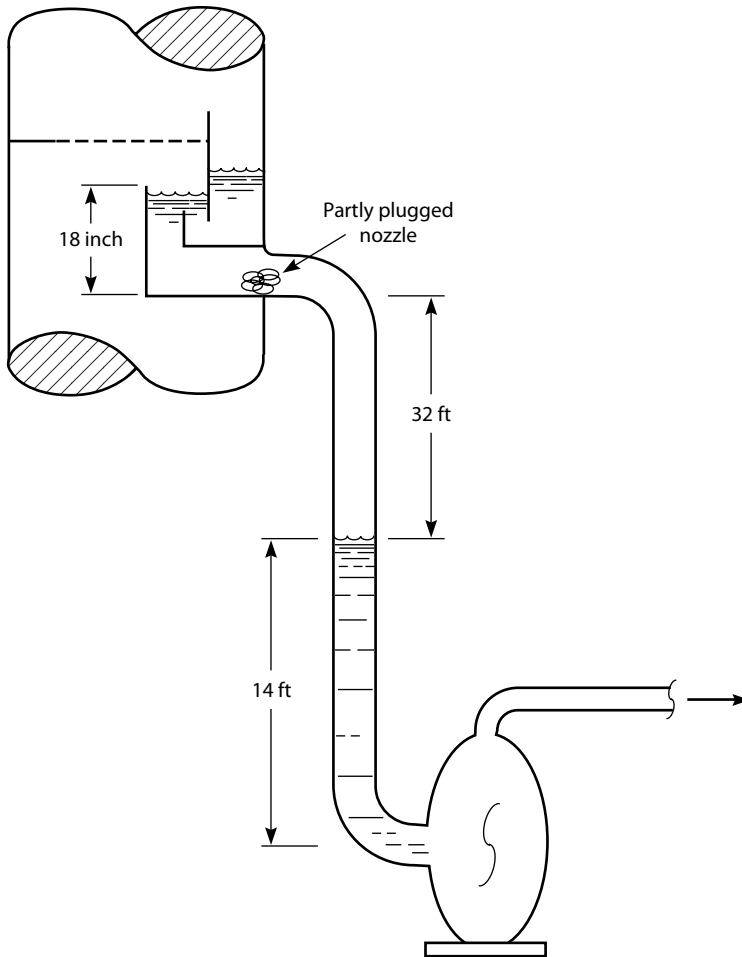


FIGURE 36.6 Most common cause of cavitation.

would cavitate or slip. The flow rate from the pump would drop, and the water level in the suction line to the pump would partially refill. The pump's NPSH requirement would then be temporarily satisfied. Normal pump operation would be restored—but only for a moment.

Of course, it may simply be that the draw-off nozzle is undersized. To determine whether this is the case, calculate the velocity (in feet per second) V through the nozzle:

$$\Delta H = 0.34(V)^2$$

where ΔH is the hydraulic head in inches of liquid required to push 110 GPM of liquid through the draw-off nozzle. In this case, ΔH is found to be 9 inch of water. Apparently, there is twice as much pressure loss through the nozzle than there should be. This indicates that the draw-off nozzle must be partly plugged.

The 0.34 coefficient shown is the conversion of potential energy to acceleration, which includes a reasonable allowance for turbulence and friction. The theoretical coefficient is about 0.18, which only includes acceleration. In my designs, I use 0.34 for nozzle exit losses, which my experience has shown represents what I actually observe in the field, plus a small safety factor.

36.1.7 Vortex Breaker

Many draw-off nozzles, especially those in the bottom of vessels, plug because of the presence of *vortex breakers*. Many designers routinely add complex vortex breakers to prevent cavitation in pumps. But vortex breakers are needed only in nozzles operating with high velocities and low liquid levels. Corrosion products, debris, and products of chemical degradation can more easily foul and restrict nozzles equipped with vortex breakers.

Lack of available NPSH may also be caused by high frictional loss in the suction piping. If this is the case, a small reduction in flow will not noticeably increase the pressure at the suction of the pump. A properly designed suction line to a centrifugal pump should have a frictional head loss of only a few feet of liquid. However, having a large-diameter suction line and a relatively small draw-off nozzle usually will lead to excessive loss of available NPSH.

36.1.8 Marginal Cavitation

On many occasions, I've noticed that large, high head pumps with a low suction pressure do not cavitate in the normal way. They do not develop an erratic discharge pressure and an erratic flow. When these pumps are marginally short of NPSH, both flow and discharge pressure remain steady. What does happen is that the pump discharge pressure falls by 50 percent (100 to 150 psi). That is, a reduction in pump suction pressure of 1 psi causes a fall in pump discharge pressure of 100 psi.

Raising the level in the vessel from which the pump is taking flow by just 2 ft dramatically increases the pump's discharge pressure. It seems as if the pump's performance jumps from an inferior performance curve to its normal performance curve (see Fig. 34.10). I learned all about this subject on a freezing, windy, and damp day in Holland. My client had thought that something was wrong with their pump, but it was really just a minor liquid level recalibration problem.

36.2 Sub-atmospheric Suction Pressure

36.2.1 Pump Suction under Vacuum

Liquids in storage tanks are almost always subcooled. This is so because otherwise the ambient vapor losses from the tank's vent would be excessive. This creates the potential for a negative pump suction pressure.

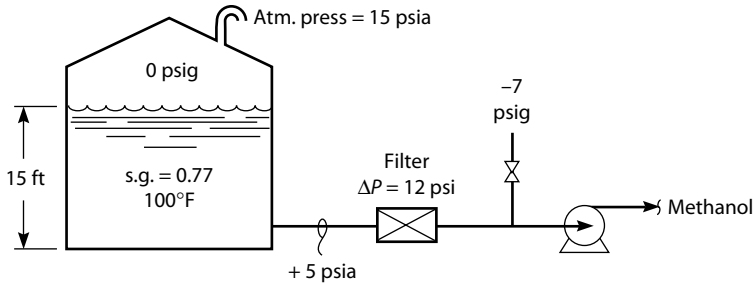


FIGURE 36.7 Pumping liquids with sub-atmospheric suction pressure.

For example, the pump shown in Fig. 36.7 is suffering from a partially plugged in-line suction filter. The positive pressure to the filter of 5 psig is due to the 15-ft head of liquid in the tank. The filter ΔP is 12 psi. Hence, the pressure at the suction of the pump is -7 psig. Since atmospheric pressure happens to be 15 psia on this particular day, the pressure at the suction of the pump is 8 psia.

The liquid being pumped is methanol, which has a vapor pressure, at the pumping temperature of 100°F, of 3 psi. The physical pressure at the suction of the pump (8 psia) minus the vapor pressure of the liquid at the suction of the pump (3 psi) equals 5 psi. To convert the 5 psi to feet of head:

$$\frac{(5 \text{ psi}) \times (2.3)}{0.77(\text{s.g.})} = 15 \text{ ft}$$

The 0.77 is the density of methanol. The 15 ft of head is the available NPSH to this pump. Does this mean that pumps may have a substantial amount of available NPSH even when their suction pressure is under a partial vacuum? Yes, if we are pumping a subcooled liquid. But this is quite common, because the liquid stored in an ordinary atmospheric-pressure storage tank is almost always well below its boiling point—that is, the liquid is subcooled.

36.2.2 Sump Pumps

The most common pump used is the sump pump shown in Fig. 36.8. The vast majority of the pumps in the world are of this type. They are the sort of pumps used to pump water out of shallow wells and from irrigation ditches. In New Orleans, we use thousands of these pumps to push rainwater over the levees and into the mighty Mississippi River.

Sump pumps can draw water up from levels as much as 30 ft below the pump's suction. But do such pumps require NPSH? Absolutely! All centrifugal pumps have some NPSH requirements. What, then, is the available NPSH to the sump pump shown in Fig. 36.8?

The physical pressure at the suction of the pump is the pressure at point A. The pump was lifting water from an oily-water sump.

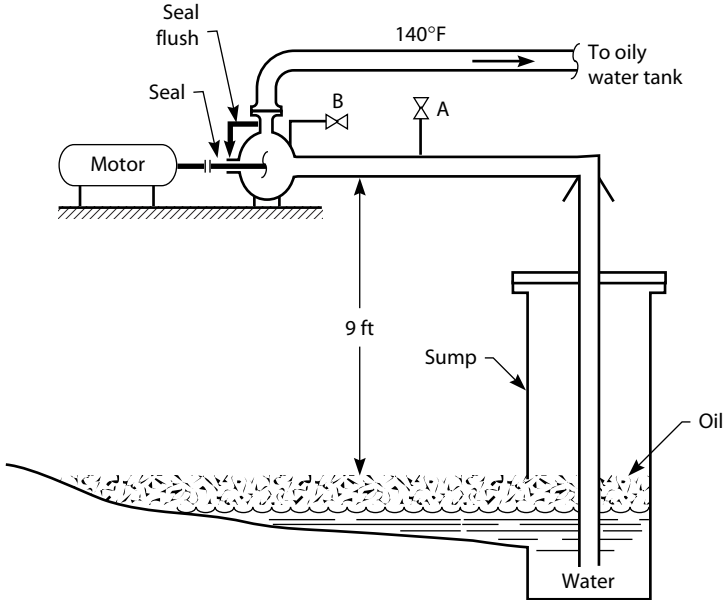


FIGURE 36.8 NPSH available to a sump pump.

The water level in the sump was 9 ft below the center line of the pump's inlet. Nine feet of water is equal to 4 psi (i.e., $9 \div 2.31$). This means that the pressure at point A was -4 psig.

Atmospheric pressure on this particular day was 14 psia. The absolute pressure at the suction of the pump was then

$$14 \text{ psia} - 4 \text{ psi} = 10 \text{ psia}$$

The vapor pressure of water at the 140°F pumping temperature is 3 psia. The suction pressure at the pump minus the vapor pressure of water at the suction of the pump is then 7 psi (i.e., 10 psia minus 3 psi).

The suction line itself was a rather rough, old cast-iron pipe, with a frictional loss of 2 psi. This frictional loss must be subtracted from the 7 psi just calculated. This leaves 5 psi available for us to convert to feet:

$$(5 \text{ psi}) \times (2.31 \text{ ft/psi}) = 11.5 \text{ ft}$$

This 11.5 ft is the NPSH available to the pump. The pump itself requires only 6 ft of NPSH to pump 1200 GPM of water. Hence, even though the pump's suction is 9 ft above the water in the sump, the available NPSH is twice the required NPSH.

A pump that is lifting very cold water (with a very low vapor pressure) through a smooth (low-frictional-loss) suction line, with a very small NPSH requirement, operating at sea level (where atmospheric

pressures are high), can lift water to its suction by perhaps 30 ft. The pump shown in Fig. 36.8 can lift 1200 GPM of water by only 15 ft. When the water level in the sump drops to 15 ft below the center line of the pump's impeller, the pump will cavitate.

Where, then, does the available NPSH to a sump pump really come from? It comes from atmospheric pressure. Atmospheric pressure at sea level is equivalent to

$$(14.7 \text{ psia}) \times (2.31 \text{ ft of water/psi}) = 34 \text{ ft of water}$$

In theory, this is the greatest height that water can be lifted from a well by a surface pump. In practice, this maximum lift height is 25 to 30 ft. Of course, water is pumped from bore-hole wells hundreds of feet deep. But this is done by submersible pumps, which are let down into the well.

36.2.3 Loss of Prime

The only real difference between a sump pump and an ordinary centrifugal pump is that the sump pump is more difficult to *prime* than the ordinary pump.

To prime an ordinary pump, we simply crack open the vent valve on top of the pump's case. The pressure of the liquid in the suction line pushes out the gas or air trapped in the pump case to the atmosphere or to the plant's flare line header. Once the pump case is full of liquid, it is primed and ready to go.

The problem with priming the pump shown in Fig. 36.8 is that the water needed to fill the pump's case (i.e., to prime the pump) is below the pump itself.

Certainly, there are self-priming pumps. These pumps are designed to compress very, very small volumes of air, and to produce very, very small discharge pressures with the air. The circulation pump on my swimming pool is of this type. After running 15 min or so, it draws through itself a few cubic feet of air before it can pick up suction. Most high-head, large-volume pumps must be primed with an external source of water. In Fig. 36.8, this is done by connecting a water hose to valve A and opening valve B. Air is pushed out of the pump case by the pressure of water in the hose.

When the pump is in operation, the suction pressure of the pump is -4 psig. This negative pressure creates a potential problem. Air may be drawn into the suction of the pump through leaks in the suction line. One especially vulnerable area is the packing gland around the valve stem of the pump's suction valve. Another potential area of air in-leakage is around the pump's mechanical seal. Air drawn into either area will cause the pump to cavitate and lose its prime. The flow through the pump will then be lost. Playing a water hose over a leaking valve stem packing gland or over a bad seal can temporarily restore the pump to normal operation. Even better, spread grease on the leaking valve stem.

36.2.4 Self-Flushed Pumps

The pump shown in Fig. 36.8 is an illustration of an installation in Port Arthur, Texas. The difficulty with this pump was that it would lose its prime after being shut down for just a few moments. What is the problem?

Answer—there is an error in the design of this pump. The error is that this is a self-flushed pump. The mechanical seals on centrifugal pumps require a lubricant or seal flush material to keep the seal faces from touching and rubbing. In most pumps, this seal flush fluid comes from the discharge of the pump itself. Such pumps are called *self-flushed pumps*.

When a self-flushed pump is running, the space between the seal faces is filled with the seal flush fluid. When the pump is shut down, the space between the seal faces is filled with the fluid in the suction of the pump. However, if the pressure at the suction of the pump is below atmospheric pressure, then air is drawn through the seal faces and into the suction of the pump. This air displaces the water in the pump's case and, with time, causes the pump to lose its prime.

To fix this problem in Port Arthur, I connected an external source of seal flush water to the pump, from a nearby washwater station. Pumps that have sub-atmospheric suction pressures, and that are not in continuous service, should not be self-flushed pumps. They should have an external source of seal flush material connected to the mechanical seal.

With the external source of seal flush, the pump no longer lost its prime when shut down. However, I then noticed an interesting phenomenon. The pump would continue to run for about an hour after it stopped raining. It produced a discharge pressure of about 100 psig. Then, quite suddenly, the discharge pressure would drop to 70 psig and the amp load on the motor driver would slip from 20 to 14 amps for its last few minutes of operation.

The cause of this odd behavior is that the water level in the sump had dropped. A layer of oil with a specific gravity of 0.70 had been drawn into the suction of the pump. The feet of head developed by the pump had not changed. But a pump's discharge pressure is proportional to

$$\Delta P = (\text{density}) \times (\text{feet of head})$$

As the specific gravity of the oil had dropped by 30 percent, so had the pump discharge pressure. The amperage load on the motor driver is also proportional to the weight of liquid pumped, which also changes with the specific gravity of the liquid.

CHAPTER 37

Centrifugal Pumps: Reducing Seal and Bearing Failures

The mechanical seal of a centrifugal pump in hydrocarbon service is the weak point for any process unit. In 1974, when I was the operating superintendent of the Amoco Oil Sulfuric Acid Alkylation Plant in Texas City, I discovered two truths about refinery operations:

1. I hated being an operating supervisor.
2. Half of my time was devoted to the repair and replacement of pump mechanical seals.

I authorized so much overtime for seal maintenance that our lead machinist purchased a new fishing boat with his extra pay. In 2013, mechanical seals are far more costly and complex, but no more mechanically reliable than in 1974. Their reliability is not a matter of their design, but of the operator's ability to operate, and most especially, to properly start up a pump. A seal may cost \$30,000 to \$40,000 to repair. Damage to such a seal is cumulative. Meaning improper start-up or operations may not suddenly cause the seal to fail, but will certainly hasten the time when it will fail.

A seal should maintain its integrity for six years, provided it is operating correctly. Most often seal life is compromised by improper start-up procedures. For example, starting up a pump without venting the seal flush lines to the inner seal on a self-flushed pump may reduce the operational seal life by six months, even though the seal faces have run dry for only a few seconds.

37.1 A Packed Pump

Centrifugal pumps were used in the 1880s. I have in my hands *Pumps and Pumping* by M. Powis Bale, published in 1889, which describes centrifugal pumps. A far more timely text, which I prefer because it

has easy to understand sketches, is *Centrifugal Pumps* by I. Karassik and Roy Carter, 1960.

The 1889 text does not mention mechanical seals, as they did not exist then. These early centrifugal pumps were packed pumps. My old shift foreman, Henry Ziprean, showed me how to replace the packing for a sour water pump during the long strike in 1974 in Texas City. I have only done this twice, 39 years ago. But, since I did it with my own hands, I've never forgotten the details.

As old Zip explained, "Mr. Norm, the purpose of the packing is to keep the liquid in the pump case from squirting out along the shaft. The packing is this square-shaped asbestos rope. Look here. I cut the rope into seven lengths. Each piece is just a bit shorter than the circumference of the shaft. I'll wrap each piece around the shaft."

"But Zip, it seems like the liquid in the pump case could squirt out through that gap between the ends of the rope."

"No, Mr. Norm. See how I placed the rope rings around the shaft? I offset those gaps by like 45°, so that the gaps don't line up. Then I'll push the rings into that cavity at the in-board [i.e., the side of the pump nearest the driver]. See how I'm stuffing the rings into the round cavity?"

"What's the cavity called, Zip? Does it have a name?" I asked.

"It's just called the stuffin' box, I reckon. Never heard no other. Next," Zip continued, "we gotta push the rings tight together inside the stuffin' box. That's done with this here lantern ring follower [lantern gland]. We tighten it up to press those seven rings of rope right up tight together."¹

"Zip, I don't think you got that lantern ring follower particularly tight. Maybe you should just pull up on those gland studs a bit more."

Old Zip smiled and said, "Nope. That ain't right. I'll just show you, Mr. Norm."

Zip switched on the pump's motor and studied the stuffing box.

"It's not leaking enough. If it don't leak some, it means those rings are packed too tight. Too much friction of the packing rings pressing on the spinning shaft. Packing will overheat. Wastes power and damages the shaft and packing. Also, need some leakage to keep that packing lubricated. See, I'll just back off some on the gland studs. Let a bit more water leak out. That looks just about right now. You can do the next one, Mr. Norm. It's that old cooling water pump on No. 1 Alky. I'll watch and you do it."

37.2 Mechanical Seal

"Zip," I asked over dinner that evening, "seems like those packing rings have to leak to cool and lubricate the inside of the stuffing box. Fine for water or even cold diesel. But not so good for butane or gasoline."

"Yep. Packed pumps have their limits. Not any good when pumping stuff that don't lubricate—like propane and butane and such. Not too safe if stuff that leaks out can catch fire easy like gasoline. Or is bad to breathe [i.e., carcinogenic] like benzene or rich amine with H_2S . Or can set up like wax. Then we gotta use a mechanical seal."

"How does that work?" I asked.

"Well," Zip answered, "those mechanical seals, they have two faces. They're made in the shape of rings. One ring is soft. It's made of carbon. The other ring is hard. It's made of tungsten-carbide. One ring is stationary. It's fixed into position on the pump case itself. It's kinda part of the pump case. The other ring rotates with the shaft and the impeller. It's kinda part of the pump's rotating assembly. Mr. Norm, those two rings or seal faces are pressed together by a strong spring, called the bellows."

"I guess, Zip, the bellows are like the lantern ring follower we adjusted with the packed pump. Adjusts the pressure on the seal faces."

"That's right, Mr. Norm. Same idea. Where these faces run past each other, that's the seal. Keeps the liquid in the pump case from running out along the shaft and escaping out into the air."

37.3 Purpose of Seal Flush

Old Zip was a drunk. He kept a bottle of Scotch in his locker. Zip came to work every day drunk, too, and I covered up for his drunken mistakes. In exchange, he shared his friendship and knowledge with me.

"Zip. If the seal faces run past each other and are pressed together by the bellows, I would think they would get real hot, just like if we overtightened the lantern ring follower on the packed pump. Why doesn't that also happen to the mechanical seal?"

"It certainly could happen, Mr. Norm, except that seal flush liquid keeps the seal cool and lubricated. The seal flush liquid flows round the outside of the seal faces, along the shaft, and back into the pump case."

"Okay. But where does the seal flush come from?"

"Out here, on our Alky Unit, most of the pumps are self-flushed pumps. That means the seal flush liquid comes from the discharge of the pump itself."

Zip sketched the drawing to explain, which I've shown in Fig. 37.1.

"Zip, it seems like you're saying that the seal itself is full of high-pressure liquid from the pump discharge. Won't that put an awful lot of pressure on the seals and cause them to leak?"

"No, Mr. Norm. You ain't lookin' at my sketch. See how that seal flush liquid comes through the restriction orifice? That orifice is just big enough for a couple GPMs to flow (one or two GPMs per inch of shaft diameter) into that seal flush chamber. The seal flush chamber is

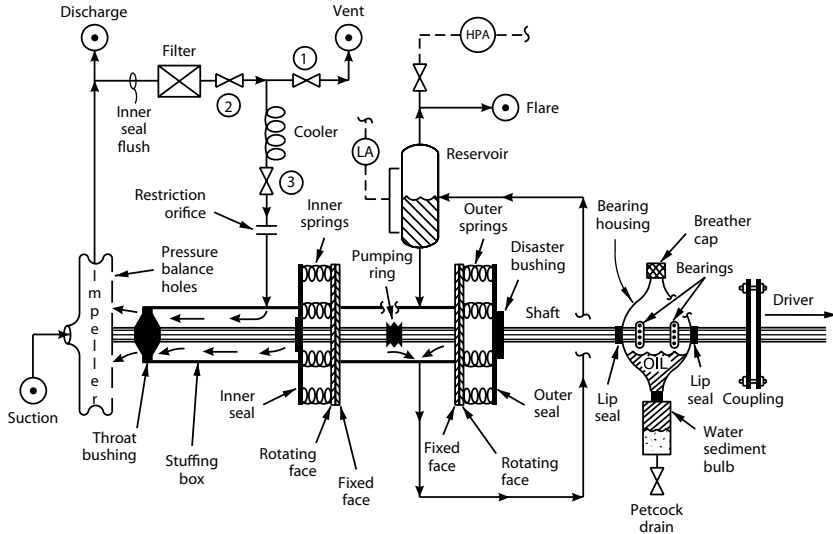


FIGURE 37.1 A double-mechanical seal. This sketch does not represent the actual mechanical construction of the seal.

like the old stuffing box on the packed pump. That little seal flush flow leaks back into the suction side of the pump case through the holes in the back of the impeller.” (See Fig. 37.1.)

“I guess, Zip, that means the seal operates just above the pressure inside the pump case, which is a little below the pump suction pressure,” I said.

37.3.1 Externally Flushed Pumps

“Zip, I’ve got another question. When I worked on our paving asphalt plant last year, did the asphalt pumps also use that black tar as seal flush?” I asked.

“No,” Zip answered. “Those were external flush pumps. Wouldn’t want to get tar between them seal faces. Those pumps were flushed with diesel oil or light gas oil. On that kind of pump, the operator needs to adjust the seal flush pressure a few psi above the pump suction pressure. Best to have a little needle valve and a rotometer. Also, you gotta have a pressure gauge after the needle valve on the seal flush line.”

“But Zip, the diesel used as the seal flush liquid will leak into the asphalt through the holes in the back of the impeller.”

“That surely is a problem. That’s why on external flush pumps, you gotta be real careful not to use too much seal flush pressure. Also, the seal flush liquid gotta be a kind that won’t ruin the product being pumped, if it contaminates a little.”

37.4 Seal Leaks

The main advantage the mechanical seal has over packing is that it's designed not to leak. If it does leak a little, then the material that escapes from a self-flushed pump is the product. For an externally flushed pump that is leaking a few GPM, the material escaping is the seal flush liquid. For an externally flushed pump mechanical seal that is blown (i.e., leaking really badly), the product being pumped is escaping along the shaft. In refineries, this is both dangerous and environmentally objectionable.

The factors that cause mechanical seals to leak are numerous:

- Loss of seal flush flow.
- Particulates in seal flush.
- Pump vibrations due to cavitation.
- Pump vibrations due to bearing damage.
- Uneven heating of pump case prior to start-up.
- Loss of pump suction pressure on pump start-up. This, in my experience, is the single biggest contribution to reduced seal life.
- Leaving a pump idle (i.e., several months) for too long.
- Ice crystals (when pumping wet propane or butane) that form between the seal faces.
- Failure to fill the seal flush line prior to starting the pump.
- Excessive seal flush pressure.

A pump seal ought to last about six years. Running a seal dry for a few seconds during start-up can reduce the seal life by six months. For a self-flushed pump that loses suction pressure on start-up, there will be little or no seal flush pressure until a normal pump suction pressure is established. (See section on starting NPSH requirements in Chap. 36.) Also, a pump that cavitates on start-up, due to the loss of suction pressure, will vibrate, which also contributes to the damage to the pump's mechanical seal faces.

Sounds of Cavitation

I have read that pump cavitation is typically accompanied by the sound of nuts and bolts ratcheting around in a metal bucket. Much of the time this is not true. I typically cannot identify a large centrifugal pump in hydrocarbon service that is cavitating by the sound of the pump. If the pump discharge pressure is erratically low, this is a more reliable indication of cavitation than sounds omitted from the pump's case. Erratically low motor driver amperage also indicates pump cavitation.

If the radial support bearings are damaged, the resulting vibrations will be transmitted to the seal and damage the seal faces. Water contamination of lube oil in the bearing house is the most common cause of bearing damage.

A pump case in hot service should be heated evenly prior to starting or switching the pump. This is usually done by the warm-up valve on the pump discharge. This valve bypasses the discharge check valve, thus permitting the hot liquid being pumped by the primary pump to back into and preheat the spare pump that is being put online.

Leaving a pump in one position for months will dry out the bearings and cause temporary distortion to the bearings and pump shaft. Running the pump for a few moments will correct these minor problems. But, during the run-in period, the vibrations may damage the seal faces. Operating experience indicates that switching pumps every week is far too frequent. Switching pumps once a year is far too infrequent. The optimum seems to be to run a pump every few months. Several smart people suggest that the primary pump should be run 90% of the time, while the spare is run 10% of the time and kept on partial standby.

To keep ice crystals from pushing the seal faces apart, due to autorefrigeration of the seal faces when pumping wet propane or butane, I used to keep a wisp of steam blowing onto the stuffing box to melt any ice crystals. I learned this nice trick from old Zip.

Particulates in the seal flush can scratch the seal faces. I've never actually encountered this problem myself, but it can be prevented by installing a Centrifix Filter on the seal flush line.

On one hydrocarbon pump, the restriction orifice was omitted from the seal flush line. This greatly increased the pressure on the seal faces. The seal blew out with a large and quite dangerous emission of hydrocarbon vapor to the atmosphere.

For self-flushed pumps, it is necessary to vent and fill the seal flush lines completely. Otherwise, on starting the pump, there will be no seal flush flow for a second or so, which will materially damage the seal.

For externally flushed pumps, this is not a problem as long as the seal flush is commissioned before the pump is switched on. Which is a problem.

37.5 Wasting External Seal Flush Oil

When a pump is not in operation, it is not necessary to have the seal flush flowing. Obviously for self-flushed pumps, this is not a problem, as the seal flush flow stops with the pump. But for externally flushed pumps, there is no sense leaving the seal flush running. For example, let's say I'm operating a Delayed Coker or Visbreaker charge pump. The seal flush flow for such a pump, all of which flows back into the pump suction when the pump is idle, would be one or two

GPM per inch of shaft diameter. For a 2-inch shaft with both inboard and outboard seals, that's 400 BSD. If the seal flush is clean gas oil, 400 BSD of gas oil will be recycled back through the coke drums or visbreaker soaker coils. At \$15 (U.S.) a barrel downgrading cost, that's \$6000 (U.S.) a day. All to no purpose. Just remember to turn the seal flush back on before restarting the pump.

In black oil service, I've seen asphalt plant operators wait a few minutes after a pump is shut down before they block-in the external seal flush flow. Of course, for autostart pumps, leaving the external seal flush closed is not an acceptable option. But, in the above example, maintain the autostart option is costing \$2 million a year in degraded product yields. I suppose that's the justification for Nitrogen Barrier Seals, as described below.

37.6 Double Mechanical Seal

A self-flushed pump with a single set of mechanical seal faces can leak the process fluid into the environment. If the process fluid is hydrocarbons, this sort of leakage is no longer permitted by the EPA. The sketch I have shown in Fig. 37.1 does not represent the actual configuration of a double mechanical seal component, but does explain the function of the individual pump internal components (see reference 2 for an excellent description of the actual arrangements of the internals for an overhung, wet sump, centrifugal pump).

The components of the pumps to the left of the inner seal faces are the same as that which I have previously described for a single mechanical seal. However, let's go through the function of each component starting at the left side of the sketch:

- Impeller—Accelerates the process liquid.
- Pressure Balancing Holes—Allows seal flush liquid that escapes past the throat bushing to flow back into the suction of the pump.
- Throat Bushing—Holds back pressure on the seal flush liquid against the inner seal faces. Or helps to control the flow of seal flush liquid, along with the restriction orifice.
- Filter—Removes particulates from the seal flush flow. Not always needed.
- Cooler—Removes heat from the seal flush flow. Not always needed.
- Restriction Orifice—Reduces the seal flush pressure to slightly above the pump suction pressure.
- Inner Seal—Prevents the inner seal flush liquid from escaping along the shaft into the space between the inner and outer seals.

- Inner Springs—Presses the rotating inner seal face up against the stationary inner seal face.
- Reservoir—Holds the outer seal flush liquid.
- Pumping Ring—Helps circulate the outer seal flush liquid.
- Disaster Bushing—In case of a catastrophic seal failure, reduces the flow of process fluid to the environment.
- Outer Seal—Prevents the reservoir liquid from escaping along the shaft into the environment.
- Outer Seal Flush—Cools and lubricates the outer seal faces.
- Bearing Housing—Holds a pair of bearings, the lube oil, and slinger rings (slinger ring omitted from sketch).
- Lip Seals—Also called dust covers or carbon seals. Retards contamination of lube oil with dirty and humid air.
- Breather Cap—Permits air accumulated in the bearing housing to escape.
- Water Sediment Bulb and Petcock—Used to drain water out of the bearing housing.

The double mechanical seal does not really improve the reliability of the pump's seal. In case the inner seal begins to fail, the level and pressure in the outer seal reservoir will increase. An alarm should sound in the control room alerting the operators of the seal failure. As the reservoir pot is vented to the flare, this should prevent small seal leaks from escaping directly into the environment.

The fluid that circulates between the two sets of seal faces is an environmentally safe liquid such as glycol, which is called the buffer fluid. It's circulated by the pumping ring. The reservoir may have a nitrogen makeup pressure control, cooling coil, and back-pressure control switch. I haven't shown these details.^{4,5} The main purpose of the double mechanical seal is to alert the operator that the inner seal has failed.

37.7 Dry Seals

One method of totally eliminating the use of externally flushed pumps using single mechanical seals is to use nitrogen barrier or dry seals. The seal faces are machined using a special technique. Nitrogen is then pressurized into the seal chamber and between the seal faces. These seals are very expensive (double the cost of conventional seals) and the feedback I get from my clients is mixed. For self-flushed pumps, nitrogen barrier seals would serve no purpose. Also, if a loss of flushing oil into the process fluid is not

uneconomic, I would stay with a conventional seal. In refineries, I would use nitrogen barrier seals in:

- Vacuum Tower Bottoms
- Coker Feed
- Visbreaker Feed
- Residual Extraction (ROSE)
- Resid FCU Feed

The amount of nitrogen actually consumed is very small, and certainly does not affect the process itself.

Perhaps one-third to one-half of my clients react with hostility when I suggest the benefits of N_2 seals, based on their experiences.

One of the problems with the use of ordinary N_2 barrier seals is that they will be ruined if a pump spins backward, even briefly. This will happen when a pump is shut down after turning on the spare pump, if the discharge check valve leaks.

The solution to this problem is to install Bidirectional N_2 Barrier Seals, which are designed to rotate in both directions. Considering that someone may leave open the warm-up valve around the check valve, when a pump is shut down, this may be a good investment.

37.8 Application of Nitrogen Barrier Seals Using Double Mechanical Seals

A dry or nitrogen barrier seal is typically used as the outer seal on a double mechanical seal (see Fig. 37.1). Double mechanical seals are used when the process fluid is environmentally objectionable or above its auto-ignition temperature. The nitrogen barrier seal replaces the reservoir fluid providing seal flush lubrication to the outer seal in the more conventional double mechanical seal.

As the inner seal for an externally flushed pump, such as a black oil (i.e., asphalt, resid), pump will continue to have the same loss of seal flush material to the process stream through the throat bushing with the conventional double mechanical seal as when the external seal is an N_2 seal, there is no real process incentive for use of the more costly N_2 seal.

Theoretically, dry seals have a longer operational life than seals using reservoir fluid to lubricate the outer seal. In practice, in my experience, this is not the case. I've accumulated a partial list of some of the problems relayed to me by the operators at the Syncrude and Suncor upgrading plants in Alberta, Canada, summarized in the following section.

It is possible to use N_2 on the inner seal and replace the external seal flush, but this is not a common practice and reportedly is quite costly.

As dry nitrogen barrier seals are about twice as expensive as a conventional seal, it's rather difficult to see the justification of the incremental cost.

37.8.1 Problems with Nitrogen Barrier Dry Seals

I was teaching a seminar for operators at the Syncrude facility last week, and they had a host of complaints about N_2 seals:

1. The nitrogen causes an idle spare pump to gas up and makes it difficult to switch pumps.
2. If the seal faces become wet with sour water, when a pump is taken out of service for maintenance, the seal may be damaged.
3. If the seal flush can be water (i.e., sour water or amine), the dry seal design serves no purpose.
4. If the pump runs backward, most (not all) nitrogen barrier seals will be ruined.
5. "Shouldn't cooling tower and other water pumps just be packed and not have any mechanical seals— N_2 or otherwise? Why then have our water pumps in this service been modified at great expense to double mechanical seals?" asked an older operator.

37.9 Steam Use in Seal Chamber

Sometimes low-pressure steam is connected to the pump's stuffing box or seal chamber. I used to think that this steam was being used in the same way that nitrogen is used in a dry seal. This is quite wrong. The steam was being used for an entirely different purpose. It's there to help keep the bellows (i.e., the spring assembly that is used to compress the seal faces) cool and clean. Hence, this steam pressure should be only a little above the pump suction pressure. In black oil services, the outside of the bellows can clog up with dirt without this flow of purge steam. Then the bellows would be unable to apply the proper pressure to the rotating seal face.

37.10 Pressure Balancing Holes

Referring to Fig. 37.1, note that the seal flush liquid enters the pump case between the rotating face (for internal assembly seals) and the back of the pump's impeller. The idea is for the relatively small flow of the seal flush liquid to flow back into the suction of the pump, and thus be returned to the process steam. (Incidentally, for external assembly seals, the fixed face is closer to the impeller. See reference 2 for details on internal vs. external assembly seals.)

For many impellers, holes have been drilled in the back face. These are called pressure balancing holes. They prevent the accumulation of pressure behind the mechanical seal. This could cause the seal pressure to exceed the pump section pressure, which would overpressure the seal faces.

One of my clients that pumps hydrocarbons contaminated with solids has observed that these pressure balancing holes often plug. This will cause the pump's seal to be damaged. This client has typically purchased mechanical seals designed for the pump's discharge pressure to avoid seal failure, when the impeller balancing holes plug. Only a few percent of my clients design seals for the pump discharge pressure, rather than follow the more conventional practice of designing the seal for the suction pressure.

37.11 Bearing Failures

The pump bearings support the weight of the shaft. Most pump bearings are ball or roller type. For larger pumps with double suctions (i.e., balanced impellers), there will be both an inboard and outboard bearing. For most of the smaller pumps (i.e., overhung) with only a single suction connection, there will be a pair of inboard bearings, both located in the bearing housing.

Any damage to the pump's bearings will cause vibrations that will be transmitted to, and damage the pump's mechanical seal faces. The usual causes of bearing damage are:

- Moisture in the lube oil
- Low lube oil level in the bearing housing
- Excessive oil level in the bearing housing

In the wet sump system of bearing lubrication, the oil level should be kept just a bit below the bottom of the bearing. There is a loose ring in the bearing housing that slings the oil around. It's called the Slinger Ring. There's a sight port on the side of the pump case to see this lube oil level.

Inside the Oiler Glass itself, there is a screw to make fine adjustments to the oil level. Within plus or minus $\frac{1}{4}$ inch, I don't think the oil level is all that critical.

Excessive oil level, like halfway up the shaft diameter, will trap heat in the bearings and cause them to overheat.

Moisture is drawn into the bearing housing due to the rotation of the shaft. This moisture will accumulate in the bearing housing and ruin the lubrication properties of the lube oil. In Fig. 37.1, I've shown a Water Sediment Bulb that I've used to drain water from the bearing housing. This is not a complete solution to water contamination of the lube oil. The oil and water will become emulsified in spite of water being drained from the bottom of the housing.

It's easy to watch this problem develop. The oil level in the oiler glass increases due to the water, rather than slowly decrease, which is normal.

37.11.1 Oiler Glass Level Discrepancies

Sometimes the oil level in the oiler cup or glass is quite high, but the real oil level in the bearing housing is too low. I can think of four factors that can cause this problem that result in pump bearing damage.

1. The oiler glass should be located "upstream" of the direction of rotation. By upstream, I mean that if you are looking at the outboard end of the pump, and the impeller is rotating clockwise, the glass should be on the right-hand side of the pump. Locating the oiler on the other (i.e., wrong) side causes the oil to be slightly pushed up into the glass.
2. The bearing housing breather cap is plugged. Air is pushed into the bearing housing by the spinning shaft. The resulting air pressure in the housing can push the oil up into the glass. This breather cap needs to be periodically cleaned.
3. The connection between the oiler glass and the bearing housing plugs.
4. Moisture from the humid air drawn into the bearing housing condenses and displaces the oil upward into the oiler glass, as discussed above.

Many years ago, working as an outside operator during a strike (i.e., I was a scab), I noted that on some of my steam turbine-driven pumps, the level of oil in the oiler glasses rose perhaps $\frac{1}{2}$ to 1 inch every day. I actually thought that in some mysterious way, that lubrication oil was being produced inside the bearing housing. Of course, it was just water.

I'm older now, and much wiser. As discussed in the following section, use of an oil mist lubrication system eliminates all of the problems with erroneous oiler glass level indications and with moisture in the lube oil.

37.11.2 Oil Mist System—Bearing Housing

My experience is only with the wet sump system for bearing lubrication. This means there is an oil level of several inches in the bearing housing. The oil level should be a fraction of an inch below the bottom face of the bearing. This level can be precisely controlled with a screw adjustment inside the oiler glass located on the side of the bearing housing.

However, I now understand that the oil mist lubrication system is superior to the wet sump system. In the oil mist method of bearing lubrication, a small stream of air containing atomized lube oil is

blown continuously into the bearing housing. There is no liquid level maintained in the bearing housing, which is kept completely drained.

This eliminates the two main problems with the more conventional wet sump lubrication system:

- Operating errors associated with improper lube oil levels are eliminated, as there is no level to maintain.
- The problem of water contamination of the lube oil due to moist air being drawn into the bearing housing by the spinning shaft and leaking bearing housing dust covers (i.e., the lip seals), is totally eliminated. Water infiltrating into the bearing housing will emulsify with the oil, and thus degrade its lubrication properties. With a totally drained bearing housing, water is immediately drained out of the bottom of the housing.

A large number of pumps can be lubricated from a centralized oil mist atomizing station. The design and construction of such a system is an additional and expensive project complication, compared to having an oiler glass on each pump. But, on the other hand, anything that reduces the probability of operator error is sure to increase unit reliability and reduce running up maintenance expenses. It is standard practice to have an automatic backup system for the oil mist generator.

37.11.3 Water Accumulation in the Bearing Housing of Turbine-Driven Pumps

Lieutenant Joe Petriceli was in command of the engine room in an aircraft carrier at the battle of Leyte Gulf in the Philippines in 1945. During the 1980 strike in Texas City, Joe taught me the following neat trick:

- A turbine driving a pump with an inboard bearing (which is normal) starts to blow steam out along its shaft, due to a defective gland seal.
- The steam impacts the inboard end of the bearing housing.
- The rate of water accumulation in the pump's bearing housing increases by an-order-of-magnitude.
- To retard the moisture accumulation in the bearing housing, one takes a hose with dry plant air and blows it across the shaft between the coupling and the pump's bearing housing to push the steam away from the bearings.

I worked for Amoco Oil from 1965 through 1981. Joe was just a sulfur plant shift foreman. But he was one of the most valuable employees Amoco had in Texas City. He was my friend and I think of him often.

Another trick that I have learned recently that also helps to extend the bearing life, if an oil sediment bulb is not used, is to employ a magnetic drain plug. This helps to collect any metal particles in the oil reservoir so as to prevent them from becoming a hazard to the bearings.⁴

37.11.4 Cloudy Lube Oil

If water is kept drained from the bearing housing using the water sediment bulb shown, bearing life can be extended. Once the oil takes on a cloudy appearance (0.1 percent water), the bearing life can be reduced by a factor of four to ten.⁵ During the 1974 strike in Texas City, I screwed the small water sediment bulbs into the ½-inch case drain on my pumps. I never actually looked for water in the bulbs. I would just drain a little fluid through the petcock to see if the lube oil was free of water. Then, if the level in the pump's oiler glass dropped a bit, I was satisfied that the connection between the glass and the bearing housing was not plugged.

For plants equipped with a lube oil mist system, the bearing housing has no liquid level, so water contamination of the lube oil cannot happen.

One thing I did not do during the 1974 strike in Texas City was clean the bearing housing breather cap. Thus, some of my pumps would develop a slight positive pressure in the bearing housing. Which pushed up the oil level in the oiler glass, and pushed down the level in the bearing housing. I remember being mystified as to why the levels in the oiler glasses would increase by themselves over a period of a few days.

Continuous water or air accumulation in the bearing housing is an indication that the lip seals (or dust covers) need to be replaced. These lip seals are not expensive and their replacement is quite simple (see Fig. 37.1).

37.12 Starting a Centrifugal Pump

The objective in starting a centrifugal pump is to minimize the damage to the pump's mechanical seal during start-up. The facilities required for this purpose are:

- Pump suction pressure gauge downstream of the pump suction strainer.
- Vent and valves to remove vapors from the seal flush system (shown as valves 1, 2, and 3 in Fig. 37.1).
- A ½-inch drain valve on the bottom of the bearing housing (or water sediment bulb).
- A high point vent on the top of the pump case.

- For larger pumps with the suction on top of the pump case, a spill-back warm-up line connected to the case is needed to evenly warm up the pump. For all hot pumps, a spill-back bypass is needed around the pump's discharge check valve for warm-up when switching pumps.

37.12.1 Preparation before Starting a Pump

1. For an externally seal-flushed pump, open the seal flush flow and adjust the seal flush pressure to the manufacturer's recommended pressure above the suction pressure. If this data is not available, use the pump suction pressure plus 10 percent of the pump's normal differential pressure. The seal flush pressure should not be close to the discharge pressure. This will damage the seal.
2. For the more common self-flushed pumps, vapors trapped inside the seal system tubing and coolers must be vented out and displaced with liquid. Otherwise, lubrication between the seal faces will be intermittent during start-up. A few seconds of running a dry seal can cause a year's worth of wear on the seal faces. Do not assume that because you have started up a pump without blowing the seal before, that no damage has been done.
3. Clean out the pump suction strainer. Check that the suction pressure gauge is operable and that the connection is not plugged. The unit engineer should have placed a red mark on this gauge showing the minimum required suction pressure to prevent cavitation on start-up.
4. For pumps that have an oiler glass (i.e., not used on the oil mist system), drain half of a glass, a few ounces of lube oil from the bearing housing. A bubble of air should rise in the oiler cup or glass and the cup level should drop a bit. Drain out all water and make sure the oil drained is clean and not cloudy. If not, damage to the bearings will cause vibration and damage to the seal faces.
5. Vent the top of the pump case to fill the case with liquid. For a self-flushed pump, starting with gas in the pump case will contaminate the seal flush tubing and cooler with vapor.
6. Open the spillback around the discharge check valve to warm the pump case. For pumps with the suction on top of the pump's case, open the connection allowing the hot spill-back flow to recirculate to the bottom of the pump. The suction valve needs to be partly open. Wait until the pump case is uniformly warm. Starting a pump that is cold or unevenly

heated will cause uneven stresses on the seal faces, or high-viscosity liquid will remain between the seal faces to shear the seal faces. This step only applies when switching pumps, and not for a complete unit start-up.

37.12.2 Start-Up

1. It takes a lot of energy to accelerate the liquid in the suction line of a pump. The longer the suction line, the more energy is needed. The energy does *not* come from the pump, but from the pressure of the liquid itself. Thus, the pressure at the pump suction will drop radically if you open the discharge valve too quickly. This will cause the liquid to vaporize inside the pump. That's cavitation, which is sure to damage the seal faces. The faster you open the discharge valve, the greater the reduction in suction pressure.
2. Opening up the discharge valve too slowly is also bad. Running a pump at an extremely low flow will cause vibration, overheat the seal, and dry out the faces.
3. So open the pump discharge valve as rapidly as possible, consistent with not permitting the suction pressure to fall below the "red mark" on the pump suction gauge, that is, minimum pressure to avoid cavitation on start-up (see Chap. 34, Sec. 34.2). Do not be guided by the pump discharge pressure. By the time the discharge pressure becomes erratic, vapor is already being generated inside the pump case with consequent damage to the seal faces.
4. Running a pump in a cavitating mode during start-up may not damage the seal faces to the extent that the seal will fail. However, a year's worth of life from a seal can be lost in one minute due to cavitation on start-up.

Improper pump start-up is like cigarette smoking. The damage is cumulative. Switching pumps every month or two is a good idea to extend seal life. But not if starting-up the idle pump is done improperly. Then, the seal life will be shortened even further. Pumps that sit idle for more than three months will cause bearings to be damaged when the pump is restarted.

37.12.3 Additional Valving for Start-Up

The valving referred to in my start-up procedure is not provided in most applications. To evenly warm-up a large pump with the suction on top, you will have to install a connection from the warm-up line to the case drain at the bottom of the pump. If the pump case is not warmed-up uniformly, this will cause the seal faces to misalign during start-up.

Referring to Fig. 37.1, in addition to valve 1, the valves shown as valves 2 and 3 must also be added. Without valves 2 and 3, the complete seal flush line cannot be filled with liquid. For externally flushed pumps, valves 2 and 3 are not required.

For pumps with double mechanical seals, the fluid level in the reservoir should be checked. The correct level is in-line with the reservoir return nozzle. Too low a level will reduce the fluid circulation rate.

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CHAPTER 38

Control Valves

I am an anticommunist. That was not always so. But working in the former Soviet Socialist Republic of Lithuania altered my views about communism. Figure 38.1, drawn to scale, illustrates the reason for my revised political orientation. Really! That's the way the piping and control valves looked in the field. It happens because of bad design practices:

1. The mechanical engineer specifies a pump so that it produces excessive head and flow, so that he cannot be accused of industrial sabotage by undersizing the pump.
2. The piping engineer oversizes process lines to avoid being branded an enemy of the people.
3. The Soviet instrument engineer must then install a small control valve that will consume all the excess pressure without operating too far in the closed position.

Control valves only work properly within a certain range of positions. When a control valve is mostly closed (perhaps 5 to 10 percent open), opening it a little bit more will increase flow a lot. When a control valve is mostly open (perhaps 80 to 90 percent), opening it a lot more will hardly increase the flow at all. To summarize, a control valve can only respond to a change in position in a linear fashion, between 20 to 80 percent open. To avoid the control valve operating in the non-linear, mostly closed position, the Soviet instrument engineer installed a small control valve that has a giant ΔP at 50 percent open.

If a control valve is operating too far closed, I will partially close the upstream gate valve to force the control valve to operate in the 20 to 80 percent range. This is a bad practice. The high velocity passing between the gate and the seat will erode the valve seat. Then, when the valve is shut, it will leak through and fail in its function as an isolation valve.

A normal installation will have the control valve one line size smaller than the process piping. For example:

- Line size = 10 inches, control valve = 8 inches
- Line size = 8 inches, control valve = 6 inches

- Line size = 6 inches, control valve = 4 inches
- Line size = 4 inches, control valve = 3 inches

Thus, simply changing the trim to a maximum size in an existing control valve may not eliminate the control valve as an hydraulic restriction. The term “trim” refers to the internals inside the control valve. If the control valve is smaller than the line size, reducing the control valve pressure drop to a minimum will be more complicated than just changing the trim. The piping reducers shown in Fig. 38.1, on either side of the control valve, will also have to be resized. This will be a big job if these piping reducers are welded rather than flanged. I made this mistake on a process revamp project in Lithuania. I specified that the maximize size trim on a control valve be installed. But I forgot to specify that a full line size control valve also be installed.

Many engineering design organizations adhere to the following practice in sizing control valves and pumps. The control valve operating at normal design conditions will consume 50 percent of the frictional losses through the system. For example, for a pumparound circuit on a distillation tower:

- Frictional loss through heat exchangers = 60 psi
- Head increase of flowing fluid is equivalent to 20 psi (about 50 ft)
- Pressure drop through the control valve = 60 psi
- Total differential pressure developed by the pump = 140 psi

While this practice may result in good process control, it also causes energy waste due to parasitic pressure loss in the control valve. It’s a conservative way to design pumping systems. But for me, it’s just communism. It’s an abdication of our responsibility to design process equipment that benefits the long-term energy efficiency of the plant.

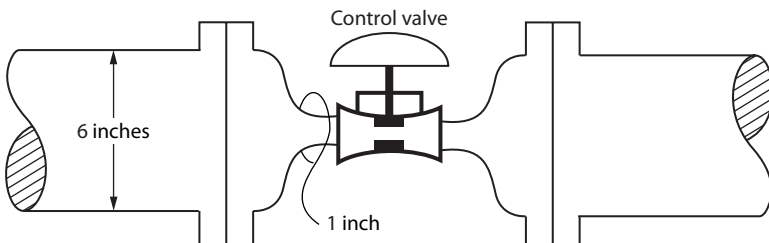


FIGURE 38.1 Over-sized pump and piping cause under-sized control valve.

38.1 Pumps and Control Valves

The relationship I'll now describe between a control valve and a pump applies to a constant speed motor-driven centrifugal pump. This covers more than 90 percent of the systems we encounter.

For variable-speed pumps, such as steam turbine-driven pumps, control valves should not be used. The facility shown in Fig. 38.2 is a control scheme that is used not infrequently at older plants. It's a fine piece of technology, which has been lost with time. No control valve is used. The turbine speed is altered to directly maintain the vessel level. With the increasing use of variable-speed alternating-current motors, elimination of parasitic control valves should become more common.

For the ordinary fixed-speed motor centrifugal pump, I suggest the following rules when selecting a control valve:

- The valve should operate between 40 and 60 percent open at design conditions.
- If I want to allow for future expansion, I'll oversize the valve and use an internal trim of reduced size.

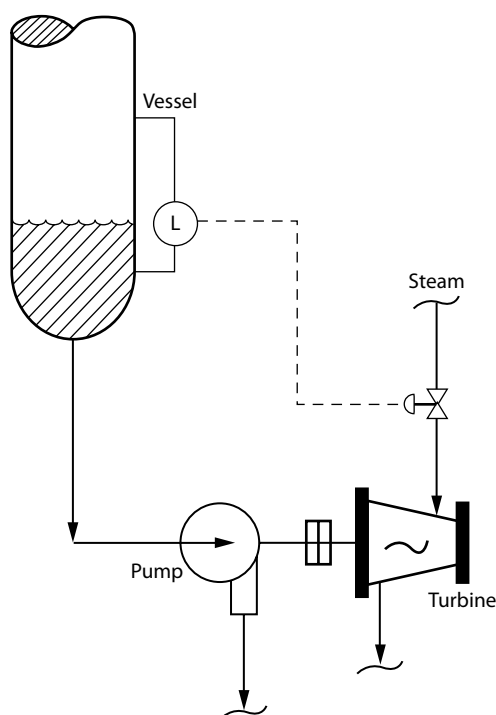


FIGURE 38.2 Variable-speed pump eliminates control valve.

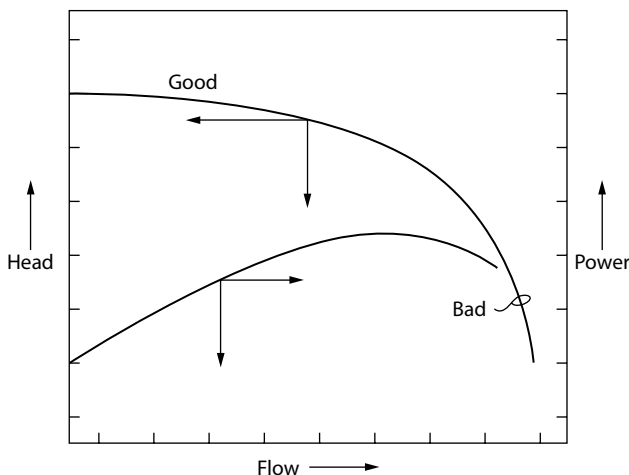


FIGURE 38.3 Centrifugal pump curve showing region of good controllability.

- The normal pressure drop across the control valve should be sufficient for the pump to operate (see Fig. 38.3) on the flat part of the pump's performance curve. These curves are published in reference manuals by the pump vendors. The flat portion of the curve in Fig. 38.3 has been labeled "good."
- The control valve should not be sized for the steep portion of the curve. I have labeled this part of the curve "bad" in Fig. 38.3.

The idea is for the centrifugal pump and control valve to work as a team to control flows. The pump and valve have to be selected together if the valve is to control the process with a minimum of wasted pumping power.

38.2 Operating on the Bad Part of the Curve

I last saw this problem at a 100,000 BSD flashed crude charge pump in Corpus Christi, Texas. It looked like the pump was cavitating. The pump discharge pressure was erratically low. The flow was also swinging. The control valve position was erratic. I checked the motor driver amps, which were also erratic. All observations indicated cavitation.

As I did not have reference to the pump curve in the field, I now proceeded as follows: I switched the control valve from automatic flow control (FRC) to manual. The flow became steady, as did the discharge pressure and motor amperage. Now I understood the problem was not cavitation, but a control problem.

As I manually closed the valve, the amperage on the motor driver went up. This meant that the pump discharge pressure was increasing faster than the flow was decreasing. Check the flow versus power curve in Fig. 38.3. The pump was operating on the steep or “bad” portion of its performance curve.

Normally, motor amps increase as a downstream control valve is opened. If you observe the opposite, then the control valve operation will be erratic, because you are attempting to control the flow too far out on the pump’s curve. This is another reason why the centrifugal pump and the control valve need to be designed to work together as a team. This is exactly what the communist engineers in Lithuania failed to do.

38.3 Control Valve Position

The control valve position shown on the control screen is not the actual control valve position. The valve position displayed in the control center is the required valve position. It is quite wrong to assume the valve position shown in the control room represents the real valve position in the field.

Liz used to make the same mistake. She would tell me to uproot a tree stump, then assume it was done. She is smarter now. She inspects the job in the field. Liz suspects I’ll cover the stump with earth and pretend that I extracted it from our garden. It’s the same in the plant. To know the real control valve position you must go into the field and inspect the valve.

Control valves have a tendency to stick. Even though the valve is shown to be moving on the panel screen, it may not be moving in the field. This is easy to check. The stem of the valve should be shiny and bright. This indicates that the stem has been traveling up and down recently. If the stem is covered with dirt, ask the panel operator to “stroke” the valve. This means he will fully open the valve and then fully close the valve. The valve stem has a position indicator. You can observe in the field if the valve opens and closes to its full extent.

Some control valves move in fits and starts. They stick and then jerk open suddenly. When I consulted in a refinery in Aruba, the fuel gas pressure was extremely erratic. High fuel gas pressure caused excess gas to be flared. Low pressure caused propane to be vaporized into the fuel gas header. Sometimes the plant would be flaring fuel gas and making up with propane simultaneously!

I worked on this problem for six years. Control experts devised advanced, real-time, computer control programs. Meetings were held to resolve the issue. Finally, I observed that the control valve on the liquid propane flow to the vaporizer would stick in a closed position and suddenly jerk open to 60 percent, where it would stick. It took six years for me to identify and two hours to correct the cause of the unstable fuel gas pressure. Of course, this erratic valve position was not seen on the control center panel.

38.4 Valve Position Dials

Figure 38.4 shows a control valve installation in the field. Next to the control valve there will be a small cabinet with three dials displayed:

- *Supply*: This is the supply air pressure. It should not change. If there is no instrument air pressure, the control valve will not work.
- *Signal*: Sometimes labeled "Input." This is a signal from either the control center (if control is remote) or from another instrument (if control is local).
- *Output*: This is the instruction to the control valve as to its required position.

The Signal and Output dials should have the same reading. Let's assume that this is an "air-to-close valve." That means that more air pressure closes the valve. Typically, the supply air pressure is 30 psig. If the air output pressure to the control valve is 3 psig, the valve should be 100 percent open. If the air output pressure to the control valve is 15 psig, the valve would be shut. We say the valve operates

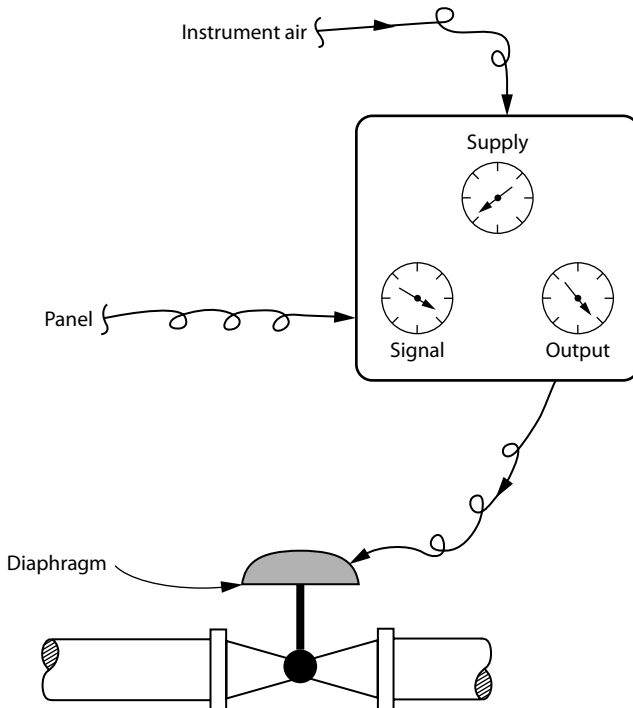


FIGURE 38.4 Air-to-close control valve.

between 3- to 15-psig air pressure. If the output air pressure was 9 psig, the valve should be 50 percent open. But if the valve is stuck, it may be in any position. It will not open regardless of air pressure to the diaphragm. Or the diaphragm itself may have malfunctioned.

38.5 Air-to-Open Valves

Many valves are air-to-open valves. The process engineer decides whether a control valve should be air-to-close or air-to-open, based on the fail-safe position of the valve if the instrument air pressure is lost. For example:

- A back-pressure control valve on a wet gas absorber should fail in an open position on loss of instrument air pressure.
- A fuel gas regulator for a control valve to a furnace should fail in the closed position on loss of instrument air pressure.

For many (but not all) control valves, you can tell if air pressure is supposed to open or close the valve. If the tubing to the diaphragm is on the top, as shown in Fig. 38.4, then the air pressure is usually (but not always) going to close the valve. When in doubt, ask the panel operator to move the valve. Touch the stem with your gloved finger and see what happens to the air pressure of the output dial.

38.6 Saving Energy in Existing Hydraulic Systems

The pumping power lost across a control valve is approximately:

$$\text{KWH} = \frac{\Delta (\text{GPM}) (\text{s.g.})}{2200}$$

where KWH = power in kilowatts converted to heat in the control valve

Δ = pressure drop, psi

GPM = (U.S.) gallons per minute

s.g. = specific gravity

Installation only of a larger control valve, or larger valve trim, will save zero energy regardless of the result of the above calculation. In addition to modification of the control valve, the upstream centrifugal pump must also be modified. Assuming that the pump is driven by a fixed-speed AC motor, the size of the pump's impeller has to be reduced. The smaller impeller plus the larger control valve will result in:

$$\% \text{ KWH} = 100 - 100 \frac{(\text{DN})^3}{(\text{DO})^3}$$

where % KWH = percent of power saved on the motor

DN = diameter of new impeller

DO = diameter of old impeller

The pump vendor publishes a family of curves for different size impellers. Using the rules described in this chapter, you can select an energy efficient combination of a new pump impeller and a new or modified control valve.

The size of the motor is not significant. The power required to run a motor depends on what it is driving, not on the horsepower rating of the motor itself.

Even on a brand new unit, you should not be surprised if 30 to 50 percent of the amperage on a driver can be saved. Again, for turbine-driven pumps the goal is not to optimize the size of the control valve but to totally eliminate the control valve.

38.7 Control Valve Bypasses

The hydraulic capacity of a system is sometimes limited by pump capacity and sometimes limited by the sizes of the control valve. To discriminate between these two problems:

1. With the control valve wide open, open the control valve bypass.
2. If the flow increases by a lot, the control valve is too small.
3. If the flow increases very little (less than 5 percent), the problem is the pump.

Operating with a control valve bypass open is not permitted by many operating companies. In theory, I agree that control valve bypasses should be kept closed. In practice, I frequently have opened bypasses. For example, we have a fired heater that is limited by the unit charge pump. Opening the bypasses around the heater pass control valves seems reasonable and safe. But suppose there is a leak in a heater tube. The panel operator closes the pass control valve remotely to stop the fire, but the flow continues through the bypass valve. The reader can imagine how I became so smart on this subject.

Running for a long period of time with a control valve bypass open in hydrocarbon service should be considered a change in operation requiring a HAZOP review. For me, as a field troubleshooter, I'll always close all control valve bypasses I opened during my investigations before I leave the plant in the evening.

38.8 Plugged Control Valves

When pumping dirty water through a small ported control valve (1 inch or less), the valve may easily plug. I can back-flush the valve to blow the dirt out (refer to Fig. 38.5):

1. Open control valve D 100 percent.
2. Close valve B.
3. Open valve C.
4. Open drain valve A. (Perhaps routing the drain via a hose to the flare or enclosed sewer might be safer.)

Water will now flow backward through the control valve. Especially since there will be little or no back-pressure at valve A, I have found this to be a highly effective method to clear small plugged control valves.

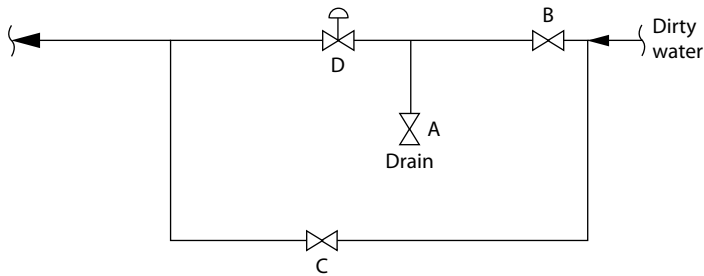


FIGURE 38.5 Clearing a plugged control valve.

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CHAPTER 39

Separators: Vapor-Hydrocarbon- Water

Liquid Settling Rates

This chapter precedes the chapter on centrifugal compressors for a sound reason. Damage to high-speed centrifugal compressors is most often associated with slugs of liquid entering the compressor from an upstream knockout drum. The other major use of knockout drums is to separate droplets of liquid from gas streams to be used for fuel or further processing. Separating droplets of water from liquid hydrocarbon is also discussed in this chapter.

39.1 Gravity Settling

The majority of the process vessels you see in your plant are gravity, vapor-liquid separators. Their main purpose is to settle out droplets of entrained liquid from the up-flowing gas. Factors that affect the settling rate of these droplets are

- *Droplet size.* Big droplets settle faster than little droplets. Contrary to that nonsense we were taught in school, cannonballs do drop faster than BBs—if one accounts for the resistance of air (BBs are small spheres of lead shot used in air guns).
- *Density of vapor.* The less dense the vapor, the faster the droplet settling rate.
- *Density of liquid.* The more dense the liquid droplets, the faster the droplet settling rate.

- *Velocity of vapor.* The slower the vertical velocity of the vapor, the faster the settling rate.
- *Viscosity of the vapor.* The lower the vapor viscosity, the faster the settling rate.

All these ideas have been put into one equation, called *Stokes' law*. Nothing against Sir Frederick Stokes, but vapor viscosities are almost always so small that they do not affect settling rates. Also, we never know the particle size distribution of the droplets. There is a more fruitful way to look at the settling tendency of droplets of liquid in an up-flowing vapor stream, as shown in Fig. 39.1. The method states

- The lifting force of the vapor is proportional to the momentum of the vapor.
- The lifting force of the vapor is inversely proportional to the density of the liquid, which we will call D_L (lb/ft³).

I'll put this in equation form:

$$\text{Tendency to entrain droplets} = \frac{\text{momentum vapor}}{D_L} \quad (39.1)$$

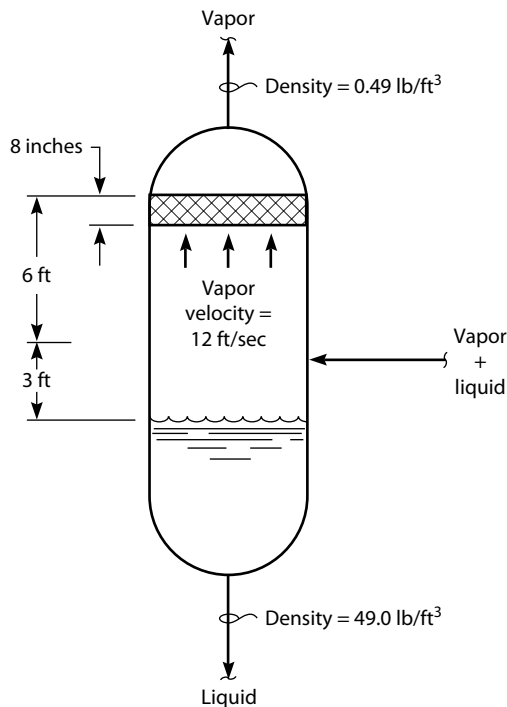


FIGURE 39.1 A vapor-liquid knockout drum.

The momentum of the vapor equals

$$M \cdot Vg \quad (39.2)$$

where M = mass of vapor, lb

Vg = velocity of vapor, ft/s

The mass flow, or weight of the vapor flow, per unit of vessel cross-sectional area is

$$M = D_v Vg \quad (39.3)$$

where D_v = density of vapor, lb/ft³.

Substituting Eqs. (39.2) and (39.3) into Eq. (39.1) gives us

$$\text{Tendency to entrain droplets of liquid} = \frac{D_v}{D_L} Vg Vg = \frac{D_v}{D_L} \cdot Vg^2 \quad (39.4)$$

Allow me now to call this tendency to entrain droplets K^2 , where K is an empirically derived number, the *entrainment coefficient*, describing the tendency to entrain droplets of a randomly produced size distribution. We then have

$$K^2 = \frac{D_v}{D_L} Vg^2 \quad (39.5)$$

or

$$Vg = K \left(\frac{D_L}{D_v} \right)^{1/2} \quad (39.6)$$

Thousands of experiments have been made both in the plant and in the laboratory to determine K values for different levels of entrainment. Here are the results:

- $K = \leq 0.15$: very slight entrainment
- $K = 0.23$: normal to low entrainment (less than 1 wt.% liquid in vapor)
- $K = 0.35$: high entrainment
- $K = \geq 0.50$: very severe entrainment (more than 5 wt.% liquid in vapor)

I feel sure, dear reader, that you are objecting to my use of such terms as "slight" and "severe," as these are qualitative terms. Well, we do not know enough about entrainment to quantify it. Also, we have neglected such obviously important factors in the knockout drum, shown in Fig. 39.1, as:

- Vertical height between the inlet nozzle and the vapor outlet
- The clearance between the liquid level (or, more properly, the froth level) and the feed inlet

- The turbulence of the vapor and liquid as they enter the drum
- The degree of dispersion of the liquid in the vapor

All I am saying is don't take Eq. (39.6) or the tabulated K values as the ultimate truth. The calculated tendency of a vapor to entrain liquid is more akin to an educated guess, rather than a precise engineering calculation.

A simple K -value calculation might be helpful here. Let's use Eq. (39.6) and Fig. 39.1 to calculate the tendency to entrain droplets of liquid in the knockout (KO) drum shown in Fig. 39.1.

$$12 = K \left(\frac{49.0}{0.49} \right)^{1/2}$$

$$K = \frac{12}{10} = 1.2$$

where the K value of 1.2 indicates that extremely severe entrainment would be certain from this KO drum. If we expected moderate entrainment, the volumetric gas flow to the vessel would have to be reduced by 80 percent, or the vessel diameter would have to be more than double.

Normal vertical knockout drums are designed for a K value of about 0.20 to 0.25. If we are installing a KO drum ahead of a reciprocating compressor—and they really hate liquids in their feed—a K value of 0.14 might be selected. If we really do not care very much about entrainment, a K value of 0.4 might be selected. An example of this would be venting waste gas to the flare from a sour-water stripper reflux drum.

39.2 Demisters

Figure 39.1 shows an 8-inch-thick demister below the top head. A *demister pad* resembles a giant Brillo pad without the soap. Many process plants have discarded demister pads lying around their scrap heaps, so you may have already seen what they look like.

The theory of operation of a demister is simple. Vapor and droplets of liquid strike the demister with a substantial velocity. The force of this impingement velocity causes the tiny droplets of liquid to coalesce into larger droplets. The heavier droplets fall out of the up-flowing vapor.

For a KO drum with a demister to work properly, it apparently must have a K value of at least 0.15 to 0.20. I say “apparently” because in at least one service, I have noted an increase in entrainment from a vessel equipped with a demister at lower K values. This service was a sulfur plant final-effluent condenser. I suppose the vapors must strike the demister's fibers with some minimum force, to encourage droplets to coalesce.

Knockout drums equipped with demisters, according to what is written in other books, can tolerate K values 30 to 50 percent higher than can ordinary KO drums. More importantly, demisters definitely do scrub out much smaller droplets from a flowing gas stream than could settle out by gravity settling alone. The most common example

of effective demister use is the removal of entrained boiler feedwater from steam generators. Boilers equipped with demisters produce steam of better quality; that is, the removal of the entrained droplets of water is accelerated and also hardness deposits are removed from the flowing steam. This is especially beneficial in keeping downstream superheater tubes and steam turbine rotor blades free of hardness deposits.

39.2.1 Demister Failure

A rather large percentage of demisters I have encountered wind up on the scrap heap. Usually, this is due to plugging. If the flowing vapor is entraining corrosion products, coke, or other particulates, the demister will probably plug. The particulates will stick to the demister's fibers. Also, I have seen terrible demister plugging due to corrosion of the demister's fibers. Because of the large surface area of the demister, a relatively low corrosion rate will produce a large amount of corrosion deposits. It is much like that rusty Brillo pad you left by the kitchen sink for a few weeks.

When a demister plugs, it increases the pressure drop of the vapor. But the pressure drop cannot increase a lot, because the demister will break. Demister failure creates two problems:

- The dislodged sections of the demister pad are blown into downstream equipment, as into the suction of a centrifugal wet-gas compressor.
- The failed demister promotes high localized velocities. Vapor blows through the open areas of the vessel. The remaining sections of the demister pad impede vapor flow. The resulting high, localized vapor velocities create more entrainment than we could have had without any demister.

Knockout drums handling gas with particulates are usually better off without a demister, unless there is a provision to back-wash the demister. The demister must be made out of a corrosion-resistant material. In many services, a high-molybdenum-content stainless steel is sufficient.

A little bit of entrainment is not always bad. For example, entraining a clean naphtha mist into the suction of a wet-gas centrifugal compressor is fine. The mist helps keep the rotor parts clean. If the steam from a boiler is to be used to reboil towers, entrainment is of no consequence, as long as the steam is not superheated.

39.3 Entrainment Due to Foam

A light, frothy foam is naturally more susceptible to entrainment than a clear, settled liquid. Knockout drums handling foam, therefore, should be designed for a lower K value. Perhaps the K value that can be tolerated might be reduced by 25 to 35 percent in the anticipation of such foam.

But foam may have a rather dramatic effect on entrainment, through a mechanism that is unrelated to vapor velocity. The problem I am referring to is high foam level. We already discussed this problem in Sec. 11.1.2, in the section devoted to level control. Figure 39.2 illustrates this problem.

The level shown in the gauge glass is 18 inches below the inlet nozzle. This is okay. But inside the drum we have foam. The average density of the foam plus liquid in the drum might be 0.60. The density of the clear liquid in the gauge glass is 0.80. Referring back to the logic in Chap. 11, the foam level in the tower is then 33 percent higher than the liquid level in the gauge glass.

Once this foam level rises above the feed inlet nozzle, the vapor in the feed blows the foam up the drum. A massive carry-over of liquid (or foam) into the vapor line then results. I mentioned above that a

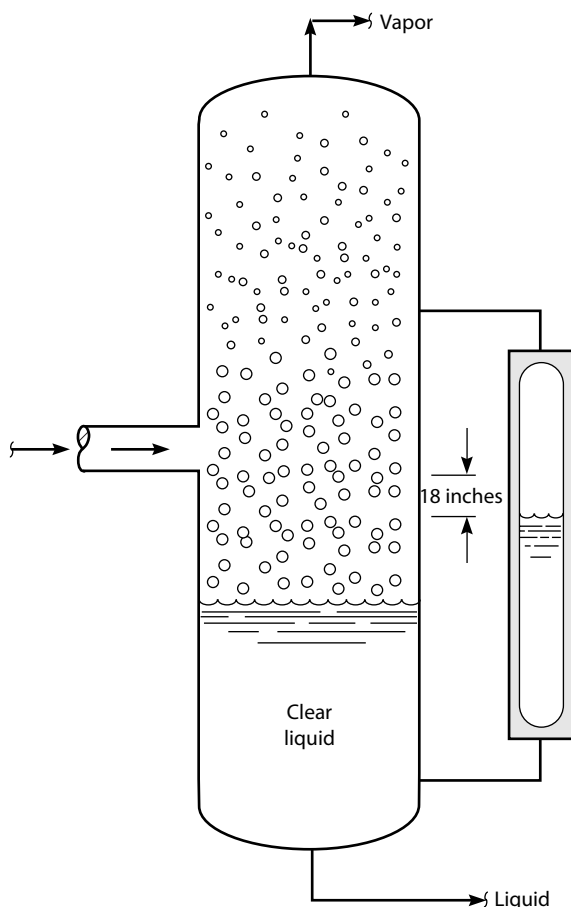


FIGURE 39.2 Carry-over of foam due to a high froth level.

small amount of mist carry-over to a centrifugal compressor helped keep the rotor clean and did no harm. Sudden slugs of liquid into the compressor inlet are another matter. They are likely to unbalance the rotor and cause substantial damage. If the compressor is a high-speed (10,000-rpm) multistage machine, it is likely to be wrecked by a slug of liquid. Even a small surge of liquid entering a reciprocating compressor can be very bad. The *intake valves* are almost certain to be damaged. Typically, the valve plate will break.

Use of Vortex Tubes marketed by EGS systems has been shown to suppress carry-over of foam.

39.4 Water-Hydrocarbon Separations

Figure 39.3 shows a reflux drum serving a distillation column. We have a reflux drum because

- The drum provides a few minutes of holdup for the overhead product and reflux. This prevents the reflux pump from losing suction should the amount of liquid discharging from the pump briefly exceed the amount of liquid draining from the condenser.

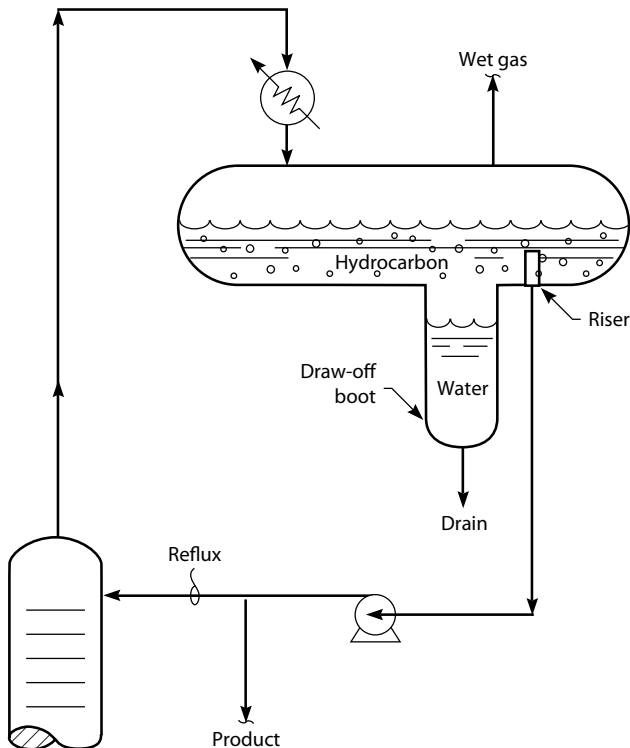


FIGURE 39.3 Water draw-off boot on a reflux drum.

- The reflux drum separates liquid and wet gas by gravity settling. A horizontal vapor-liquid separator works in much the same way as the vertical KO drum.
- The horizontal reflux drum also separates hydrocarbon liquid from water.

Why do we usually want to separate water from the reflux stream, as shown in Fig. 39.3? Some of the bad things that happen to the distillation tower if water persistently entrains into the reflux are

- The distillation tray efficiency is reduced. The water may settle out beneath the hydrocarbon liquid on the tray. This reduces contact between the up-flowing vapor and the down-flowing, internal reflux.
- The tray may flood. Water and hydrocarbon mixing on the tray deck, stirred up by the flowing gas, creates an *emulsion*. The emulsion does not separate as readily as clear liquid from the gas. Premature downcomer backup followed by tray deck flooding result.
- Tray deck, downcomer, and vessel wall corrosion is increased. Water conducts electricity a thousand times better than do liquid hydrocarbons. Corrosion involves the transfer of electrons between steel and inorganic molecules, such as
 - Hydrochloric acid to form ferric chlorides
 - Hydrogen cyanide to form ferric cyanide
 - Carbon dioxide to form iron carbonate
 - Hydrogen sulfide to form iron bisulfide

The water acts as a highway for the electrons (the term *electrons* comes from the word *electricity*) to move between these potentially corrosive molecules and the vessel wall.

Water in reflux tends to get trapped in the tower if the tower-bottom temperature is above the boiling point of water at the tower's operating pressure. The water trickles down the tower and re-vaporizes off of the hot reboiler tubes. As the water may be saturated with corrosive salts and gases, reboiler tube corrosion can be rapid. In almost every petroleum refinery service, refluxing water is a quick route to reboiler tube leaks.

39.4.1 Water Settling and Viscosity

Water droplets settle out of a lighter liquid hydrocarbon phase because of gravity. The water is denser than the liquid hydrocarbons. Liquid droplets settle out of a lighter gas phase faster because of gravity. What is the difference? The difference is viscosity. We neglected the effect on the gas viscosity when calculating the K value in KO drums. Gas viscosities are almost always very low.

But liquid viscosities are extremely variable. At a constant viscosity, the settling rate of water in another liquid due to gravity is proportional to

$$(\text{Density of water}) - (\text{density of liquid in the reflux drum})$$

For water settling out of gasoline, in the reflux drum shown in Fig. 39.3, the settling rate is between 0.5 and 1 ft/min. If a small droplet of water has to fall through a layer of hydrocarbon 3 ft deep, it would take about 3 to 6 minutes. If the hydrocarbon is less dense and less viscous than gasoline (like butane), the settling rate will be faster. If the hydrocarbon is more dense and more viscous than gasoline (like diesel oil), settling time will be longer.

The most important feature of the reflux drum shown in Fig. 39.3 is the *riser*, a piece of pipe 4 to 12 inches high protruding from the bottom of the drum. Water first settles to the bottom of the drum. The water then runs along the bottom of the drum into the water *draw-off boot*.

Sometimes corrosion products fill the bottom of the drum. Sometimes holes corrode through the base of the riser or the riser falls off. Either way, settled water can be drawn off the reflux drum into the suction of the reflux pump.

39.4.2 Interface-Level Control

Water is drained off the boot to maintain an interface level between water and hydrocarbon in the boot. These interface level controllers, especially on reflux drum water draw-off boots, are typically set locally. The operator finds the visible interface level in the boot's gauge glass, and then sets the local interface level controller to hold the level halfway down the boot. Level taps in the lower portion of any vessel tend to plug. This is especially true with the level taps in water draw-off boots. You need to blow out these taps on a frequent basis, because a high boot interface level can push water into the tower's reflux pump.

39.5 Electrically Accelerated Water Coalescing

39.5.1 Electrostatic Precipitation

One really good method to speed up the settling rate of water is with electricity. Water is a *polar molecule*, meaning that one side of the water molecule is positively charged and the other side is negatively charged. It is this polar characteristic of water that makes it a good conductor of electricity (a highway for electrons). For example, we have the story of the operator who encountered a small electrical fire and, not having any water handy, decided to extinguish the fire by urinating on it. That was the last decision that operator ever made.

Hydrocarbons, on the other hand, are nonpolar molecules. They are not particularly affected by electricity. We can take advantage of this difference between polar water and nonpolar hydrocarbons to accelerate water settling.

An electric precipitator contains sets of electric plates. A high-voltage (20,000-V) electric current is applied across these plates, or electrodes. The droplets of water are electrically attracted to these plates, or grids. The water droplets coalesce into larger heavier droplets on the plates. They fall rapidly to the bottom of the electric precipitator vessel.

These precipitators may increase the settling rate of water by a factor of about 5. For example, settling droplets of sulfuric acid out of alkylate (a light hydrocarbon) by gravity alone might take an hour. A good electric precipitator can do a better job in just 10 min. Precipitators do not really consume much electricity. It is true that the electricity is applied at a very high voltage to speed up settling, but the amount of current drawn—or rather the amount of current that is *supposed* to be drawn—is nil.

If you see the amps on your precipitator creeping up or spiking up, something is beginning to short-circuit the electric grids or insulators. Perhaps corrosion products are falling off the walls of the precipitator vessel. In my experience, this is the most common cause of precipitator failure.

39.5.2 Electric Precipitators in Mist Removal

On my sulfuric acid production unit in Texas City, we had an electrostatic precipitator to remove a liquid sulfuric acid mist from a flowing gas stream. It worked in the same way as a precipitator in liquid-liquid service. However, the electrodes or grids were not parallel plates. As illustrated in Fig. 39.4, the grids were lead tubes and lead-coated wires. The gas flowed through about 400 8-inch diameter lead tubes, arranged in parallel. It was rather like a single-pass, shell-and-tube heat exchanger (see Chap. 26).

This electric sulfuric acid–mist precipitator worked fine, until one of the lead-coated wires inside the tubes broke. Then the entire precipitator would arc. This means the amps would jump up and down, as the single broken wire flapped around inside the tube. I do not know why, but the wires failed only on weekends. Then your fearless author would crawl inside the precipitator, cut out the broken wire, and plug the tube with a wooden plug. I've saved a piece of the lead wire. It's a memento of what I do best.

39.6 Static Coalescers

A *coalescer* works in the same way as a demister, except that it is used to accelerate the removal of droplets of a heavier liquid from a flowing lighter liquid. An ordinary coalescer is shown in Fig. 39.5.

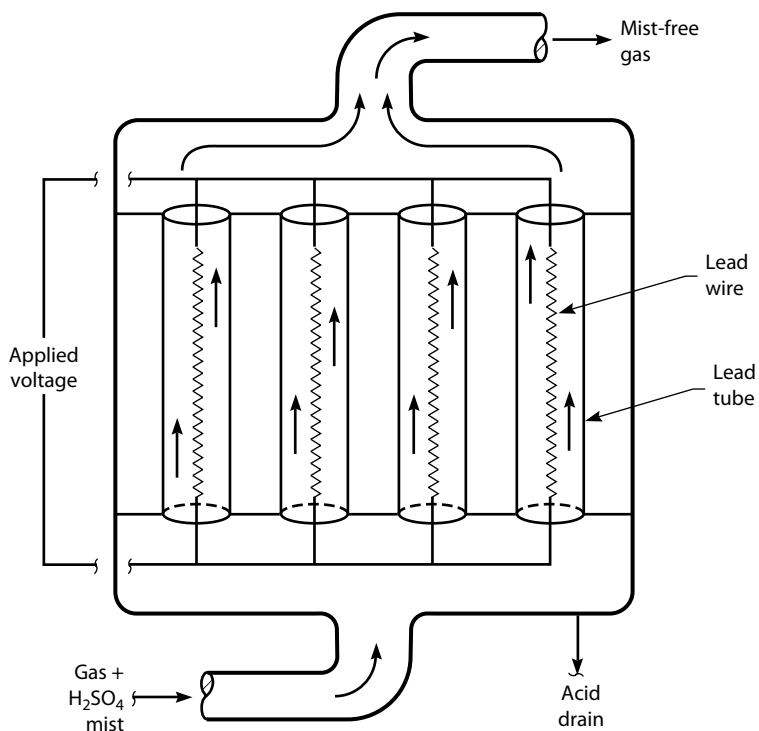


FIGURE 39.4 A sulfuric acid-mist electrical precipitator.

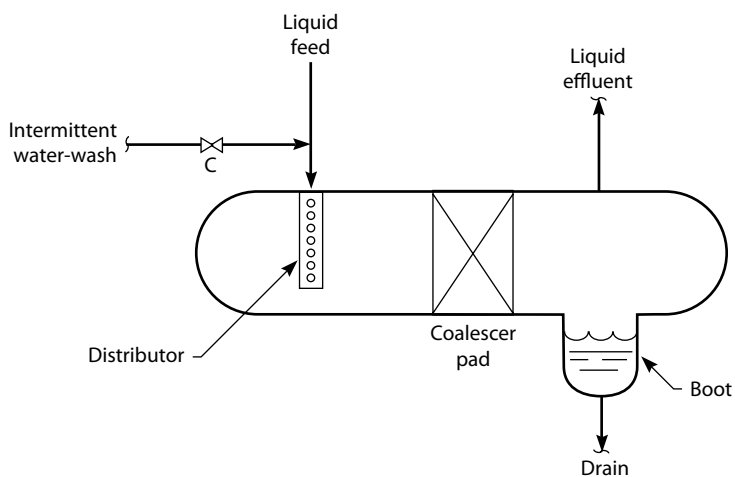


FIGURE 39.5 Horizontal coalescer.

This coalescer was used to remove entrained caustic from a flowing isobutane stream. The liquid isobutane would impact the coalescer pad at a velocity of 1 to 2 ft/min. The droplets of caustic, which have a higher surface tension than isobutane, would adhere to the surface of the coalescer fibers. As the caustic droplets grew bigger and heavier, they would drain down the fibers of the pad and into the boot.

This particular coalescer pad was made out of ordinary straw. It would gradually plug and lose its effectiveness every few days. To restore its efficiency, we would open the intermittent water-wash valve, shown in Fig. 39.5, for 15 minutes to wash off the offending deposits.

Well, one day (this was in 1974), I decided that the use of straw in a modern industrial process plant was inappropriate. I ordered a new coalescer pad constructed from state-of-the-art synthetic fibers. We replaced the sorry-looking straw pad with the really attractive, modern fiber pad—which did not coalesce nearly as well as the straw. The moral of this story is, of course, “If it ain’t broke, don’t fix it.”

Some coalescers consist of sets of parallel, slightly sloping, horizontal plates. The idea is to reduce the settling height and hence speed settling. My experience with this sort of coalescer is quite positive.

CHAPTER 40

Gas Compression: The Basic Idea

The Second Law of Thermodynamics Made Easy

This chapter establishes the basis for the Second Law of Thermodynamics. It is not critical that you read this chapter to be able to understand the more practical chapters on compression that follow. But for those readers who have technical training, wouldn't it be lovely to actually understand the basis for the Second Law of Thermodynamics. Wouldn't it be grand to really see the beauty and simplicity of the basis for the adiabatic compression work equation:

$$\text{Work} = P_1 V_1 \frac{K}{K-1} \left[\left(\frac{P_2}{P_1} \right)^{(K-1)/K} - 1 \right]$$

I have also written this chapter so that the nontechnical reader can easily comprehend the basis for this Second Law.

40.1 Relationship between Heat and Work

Julius Robert Mayer was a physician practicing near Bavaria in the 1840s. As part of his research into human metabolism, he decided to determine the equivalence between heat and work.

Heat means British thermal units, or the amount of fuel we have to burn to increase the temperature of a pound of water by one degree Fahrenheit. *Work* means foot-pounds, or the amount of effort needed to raise a one-pound brick by one foot.

The experiments that people like Dr. Mayer performed established the technical basis for the Industrial Revolution. Dr. Mayer himself laid the foundation for the main pillar supporting this technical basis.

This was the science of thermodynamics. But in the nineteenth century, they had not coined the word *thermodynamics*. They called it “heat in motion.”¹ The branch of science that we now call thermodynamics was developed by simply heating air under different conditions.

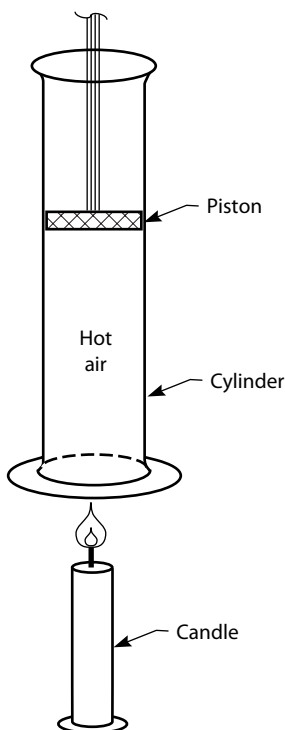
For example, let us pretend we are heating air with a wax candle. The air is confined inside a glass cylinder. We can assume that all the heat generated by burning the wax is absorbed by the air inside the cylinder. This is called an *adiabatic process*. I have shown a picture of the cylinder in Fig. 40.1.

The air in this cylinder is confined by a glass piston. The edges of the piston have been greased, so that the piston can glide without friction, up and down through the cylinder. But the edges of the piston itself have been so carefully ground that no air can slip between the greased piston and the walls of the cylinder. This means that the pounds of air contained in the cylinder below the piston will always be constant.

Dr. Mayer first wanted to determine how much wax he had to burn to heat the air inside the cylinder by 100°F. At this point in the experiment, he had to make a decision. Should he allow the expanding hot air to push the piston up? Or should he fix the position of the piston?

If the piston was kept in a fixed position, the pounds of air trapped inside the cylinder would continue to occupy a constant volume, as

FIGURE 40.1
Measuring specific
heats.



the air was heated. The pressure of the air would increase as it was heated, but the volume of air would remain constant.

If the piston was not kept in a fixed position, the pounds of air trapped inside the cylinder would stay at a constant pressure as the air was heated. The volume of the air would increase as it was heated, but the pressure of the air would remain constant.

Which is the correct way to conduct this experiment? Again, the objective of this experiment is to see how much wax has to be burned (which I will now call *heat*) to increase the temperature of a fixed weight of air by 100°F. Well, Dr. Mayer decided to conduct the experiment in both ways, to see if it made any difference. And this decision by Dr. Mayer was a turning point in human history.

Dr. Mayer already knew the weight of wax needed to heat 1 lb of water by 1°F. His British colleagues had previously determined this quantity and had called it a *British thermal unit* (Btu).

He began by heating the air with the piston in a fixed position. Thus, the volume of air heated was kept constant. The amount of heat (in Btu) needed to heat a fixed weight of air, under constant volume conditions, he called C_v . This is now called the *specific heat of air at constant volume*.

Next, Dr. Mayer heated the air, but allowed the piston to rise as the hot air expanded. This kept the pressure in the cylinder just a little bit above atmospheric pressure. The Btu needed to heat a fixed weight of air under constant pressure conditions, he called C_p .

Well, C_p turned out to be a lot higher than C_v . Dr. Mayer thought about this and concluded that he had made a mistake in his experiment. The mistake was the weight of the piston. When he measured C_p he had forgotten about the work needed to raise the heavy piston. Some of the heat generated by the burning wax was being converted to work to raise the heavy piston (work equals foot-pounds).

So Dr. Mayer repeated his experiment. He made the piston so light that its weight could be neglected. This helped, but still, C_p —now known as the specific heat of air at constant pressure—persisted in being about 40 percent larger than C_v . He reasoned that the expanding air must still be doing work and, therefore, converting some of the heat from the candle into work. But if the piston no longer had any significant weight, what sort of work could the expanding air be doing? Dr. Mayer's answer to this question changed history.

The expanding air was doing *compression work*. But what was being compressed? Not the air inside the cylinder, which, according to Dr. Mayer, was doing the work. This air was kept at a constant pressure. Certainly, this air was not being compressed. No, dear reader, Dr. Mayer said that it was the air outside the cylinder that was being compressed. In other words, all the other molecules in the sea of air surrounding our planet were pushed a little closer together by the expanding air in the cylinder.

Or, the air, which expands on heating, does work on its surrounding environment. But if we heat the air and don't allow it to expand,

it cannot do any work on its environment. Let's just stop for a moment and give all these words some real teeth.

Let's say I allow air to expand, but I do not supply it with any extra heat from an outside source. The air is expanding because I am allowing it to push away a piston in a cylinder. What do you think happens to the temperature of this expanding air? Isn't it true that whenever a gas (like air and natural gas) expands, it gets colder?

40.2 Compression Work ($C_p - C_v$)

The specific heat at constant pressure (i.e., C_p) is a measure of the amount of heat we put into the air, trapped inside the cylinder as shown in Fig. 40.1. Some of this heat is used to increase the temperature of the trapped air by 1°F. The rest of the heat goes into the work required to force the piston up and hence compressing the air surrounding the planet Earth.

The specific heat at constant volume (i.e., C_v) is a measure of the amount of heat we put into the air trapped inside the cylinder. All this heat goes to increasing the temperature of the trapped air by 1°F. None of the heat goes into compression work, because the piston remains fixed.

The difference between C_p and C_v is then *compression work*:

$$\text{Work} = C_p - C_v$$

How can we determine how much work is being done? There are two ways to calculate the amount of compression work that the piston is doing on the atmosphere of air surrounding the planet:

Method 1. Take the pounds of wax burned when C_p was measured. Then take the pounds of wax burned when C_v was measured. Find the difference between the two weights of wax. Burning 1 pound of wax generates about 18,000 Btu/lb. Let's say we burned 8 lb of wax when measuring C_p . We burned 6 lb of wax when measuring C_v . The amount of heat that has then gone into work is

$$(8 \text{ lb} - 6 \text{ lb}) \times 18,000 = 36,000 \text{ Btu}$$

or

$$C_p - C_v = \text{work}$$

We currently know that 1 Btu is equal to 740 ft-lb worth of work. Therefore, the work performed by the expanding air equals

$$740 \times 36,000 = 26,700,000 \text{ ft-lb}$$

If none of this work was wasted, we could use it to lift a 1000-lb rock up to the top of Mount Everest (which is about 26,000 ft high). But in the 1840s, Julius Robert Mayer did not know the conversion

factor of heat equivalent to work of 740 ft-lb per Btu. It had not been determined because Dr. Mayer was the one who first discovered it. So Dr. Mayer had to use the following method.

Method 2. Let's say the diameter of the piston in Fig. 40.1 was 40 ft. The area of such a piston is 1256 ft². The burning wax is causing the weightless piston to be pushed up 100 ft by the air, as it is heated and expands. The piston is being pushed up against an atmospheric pressure of 14.7 psia. Let us remember that there are 144 inches² in a square foot. So that I can say that atmospheric pressure is actually

$$14.7 \times 144 = 2117 \text{ lb/ft}^2$$

The total force of the atmospheric pressure pressing down on my piston is then

$$2117 \text{ lb/ft}^2 \times 1256 \text{ ft}^2 = 267,000 \text{ lb of force}$$

Well, work equals force times distance. The piston is traveling a distance of 100 ft. Therefore, the work done by the expanding air is

$$100 \text{ ft} \times 267,000 \text{ lb} = 26,700,000 \text{ ft-lb}$$

$$\text{Distance} \times \text{force} = \text{work}$$

Dr. Mayer used our second method. He knew that the heat of combustion of 2 lb of wax was 36,000 Btu. He divided

$$\frac{26,700,000 \text{ ft-lb}}{36,000 \text{ Btu}} = 740 \frac{\text{ft-lb}}{\text{Btu}}$$

to obtain the heat equivalent of work. It would be impossible today to design an industrial process unit without knowing this fact.

The difference $C_p - C_v$ is proportional to the amount of work the piston could perform when supplied with a total amount of heat, proportional to C_p . Then the ratio

$$\frac{C_p - C_v}{C_p} = \frac{C_p}{C_p} - \frac{C_v}{C_p} = 1 - \frac{C_v}{C_p}$$

is equal to the fraction of useful work we could recover from a total heat input proportional to C_p .

The term C_p/C_v is usually called K (the ratio of the specific heats). If I substituted K into the preceding equation, I would obtain

$$1 - \frac{1}{K} = \frac{K}{K} - \frac{1}{K} = \frac{K-1}{K}$$

Does that look familiar? It really ought to, if you have any type of engineering training. Remember the formula for compression work given at the start of this chapter and in our thermodynamics textbooks:

$$\text{Work} = P_1 V_1 \frac{K}{K-1} \left[\left(\frac{P_2}{P_1} \right)^{(K-1)/K} - 1 \right]$$

where P_1 = suction pressure, psia

P_2 = discharge pressure, psia

V_1 = suction volume, ft³/second

$K = C_p/C_v$

There is another, nonmathematical, way to think about the difference between C_p and C_v . To extract work from any process requires an energy input, like burning wax. Some of this energy will be extracted as work ($C_p - C_v$). But most of this energy input will wind up as heat (C_v).

This is the nasty, but inescapable, lesson learned from the Second Law of Thermodynamics, which is derived from Julius Robert Mayer's work. A few examples will suffice to close this chapter:

- A modern car engine converts about 12 to 15 percent of the energy in the gasoline to shaft horsepower extracted from the engine.
- The most efficient engine ever made is the Rolls-Royce high-bypass jet engine that we see on commercial aircraft. This engine converts about 39 percent of the energy in the jet fuel (which is kerosene) to thrust.
- A modern power station converts about 30 percent of the energy in the fuel burned to exported electrical energy.
- The ratio of $(C_p - C_v)/C_p$ for air is about 30 percent.

Reference

1. J. Tyndall, *Heat a Mode of Motion*, 6th Edition (Longmans, London, UK, 1880).

CHAPTER 41

Centrifugal Compressors and Surge

Overamping the Motor Driver

Have you ever heard a 12,000-hp, 9000-rpm, multistage centrifugal compressor go into *surge*? The periodic, deep-throated roar emitted by the surging compressor is just plain scary. Machines, quite obviously, are not intended to make such sounds. But what causes surge?

Another question: What happens to the amperage load on a motor-driven centrifugal compressor when the molecular weight of the gas increases? I ask this question in the following context:

- The compressor is a fixed-speed machine, as shown in Fig. 41.1.
- The suction pressure P_1 is constant.
- The discharge pressure P_2 is constant.
- The number of moles of gas compressed or the *standard cubic feet per hour* (SCFH) is constant.
- The suction temperature is constant.

We ought to be able to answer this question with Julius Robert Mayer's equation—also called the *Second Law of Thermodynamics* (see Chap. 40), which states that motor amperage (or electrical work) is proportional to

$$\text{Compression work} = NT_1 \frac{K}{K-1} \left[\left(\frac{P_2}{P_1} \right)^{(K-1)/K} - 1 \right]$$

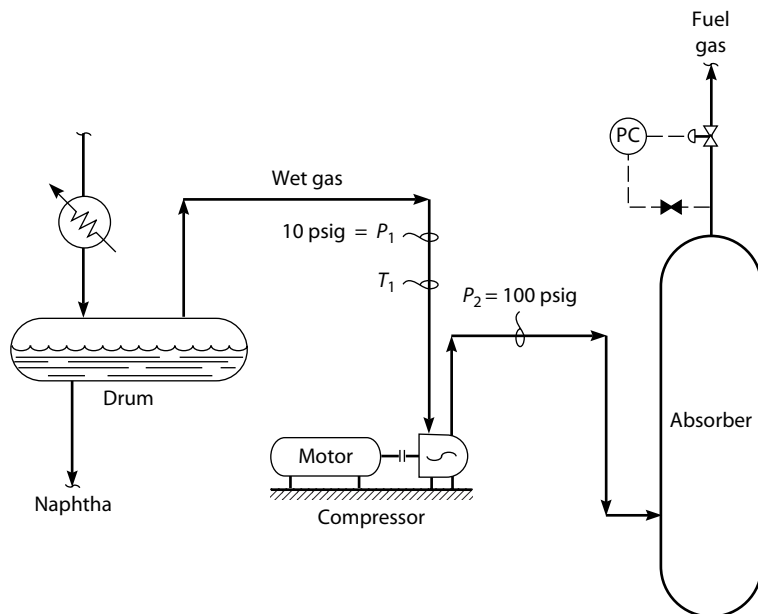


FIGURE 41.1 A motor-driven, fixed-speed centrifugal compressor.

where N = number of moles, a constant

T_1 = suction temperature, a constant

P_2 = discharge pressure, a constant

P_1 = suction pressure, a constant

K = ratio of the specific heats, C_p/C_v

We will assume that over the ranges of molecular weights we are working with that the ratio of the specific heats K is constant. This is not quite true, but this approximation will not invalidate the following statement: *According to the Second Law of Thermodynamics, as the molecular weight of the gas compressed increases, the amperage (amp) load on the motor should remain constant.*

The only problem with this statement is that it contradicts reality. When we actually increase the molecular weight of a gas, the amp load on the centrifugal wet-gas compressor shown in Fig. 41.1 does increase. This seems to contradict the Second Law of Thermodynamics. But the Second Law has never been shown to be wrong. So we have a conflict. Our experience tells us that the amp load on the motor must increase as the molecular weight of the gas increases. But the Second Law of Thermodynamics tells us that the amp load on the motor must remain the same as the molecular weight of the gas increases.

The resolution of this conflict between theory and practice, and the question "What causes surge?" will require the rest of this chapter to answer.

41.1 Centrifugal Compression and Surge

41.1.1 Mechanically, What Is Surge?

What is actually happening inside a compressor when it begins to make that surging sound? Let us refer to Fig. 41.2. When a compressor starts to surge, the gas flows backward through the *rotating assembly* (i.e., the *rotor*). This reversal of flow pushes the rotor backward. The rotor slides backward along its radial bearings. The radial bearings support the weight of the rotor.

The end of the rotor's shaft now slams into the thrust bearing. The thrust bearing constrains the axial (i.e., horizontal) movement of the rotor. Each time you hear the compressor surge, the rotor is making one round trip across its radial bearings. Each time the rotor surges, the force of the end of the shaft impacting the thrust bearing causes the thrust bearing to deform. As the thrust bearing deforms, the axial movement of the rotor increases. The spinning wheels of the rotor come closer and closer to the stationary elements (called the *labyrinth seals*) of the compressor, which are fixed inside the compressor case.

When a spinning wheel (with a wheel tip velocity of perhaps 600 miles an hour) touches a stationary element, the compressor internals are wrecked. Pieces of the wheel have been known to tear through the compressor case and kill operators. Older (1960s), lower-speed compressors seem to withstand the destructive forces of surge better than do newer, higher-speed models.

41.1.2 How Do Centrifugal Compressors Work?

Centrifugal compressors and centrifugal pumps work on the same principle. If you have neglected to read Chap. 34, "Centrifugal Pumps: Fundamentals of Operation," this would be a good time to read it. Both centrifugal compressors and centrifugal pumps are *dynamic machines*, meaning that they convert velocity into feet of head.

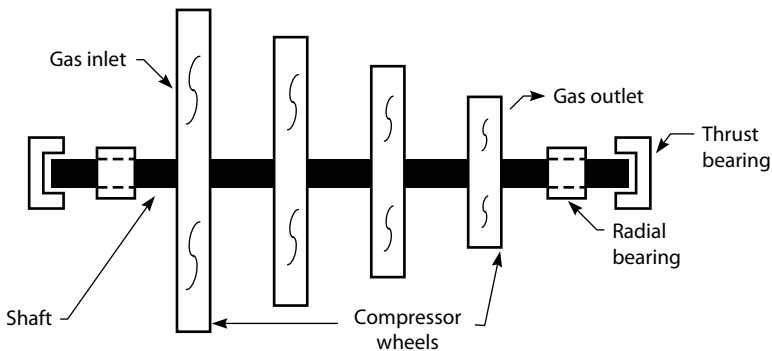


FIGURE 41.2 Rotating assembly for a centrifugal compressor.

The gas enters the compressor's rotor through the large wheel shown in Fig. 41.2. The purpose of this wheel is to increase the velocity or kinetic energy of the gas. After the high-velocity gas escapes from the vanes in the wheel, the gas enters the stationary elements fixed to the inner wall of the compressor case. This is called the *stator*. Inside the stator, the velocity or kinetic energy of the gas is converted to polytropic feet of head, or potential energy.

Brave reader, do not be afraid of the term *polytropic feet of head*. It really has the same simple meaning as described in Chap. 34, except the term *polytropic feet of head* means feet of head for a compressible fluid that is changing temperature.

To convert from polytropic feet of head to ΔP , which is really what process people are interested in, we use the following very rough approximation:

$$\Delta P \approx D_v \cdot H_p \quad (41.1)$$

where ΔP = discharge pressure minus the suction pressure

D_v = density of the vapor at the suction of the compressor

H_p = polytropic feet of head

Centrifugal compressors operate on a curve, just like centrifugal pumps. A typical curve is shown in Fig. 41.3. The horizontal axis is *actual cubic feet per minute* (ACFM). This is analogous to GPM, used on the horizontal (x) axis of centrifugal pump curves. The vertical axis is H_p (polytropic feet of head). This is analogous to the feet of head used on the vertical (y) axis of the centrifugal pump curve.

The centrifugal compressor, unless it is dirty or mechanically defective, has to operate on its curve. As the compressor discharge pressure increases, then H_p , the feet of polytropic head required, must also increase. Also, as can be seen from the compressor curve, the volume of gas compressed (ACFM) must decrease. When the volume of gas drops below a critical flow, the compressor will be backed up to its surge point.

FIGURE 41.3
A centrifugal
compressor
operating curve.

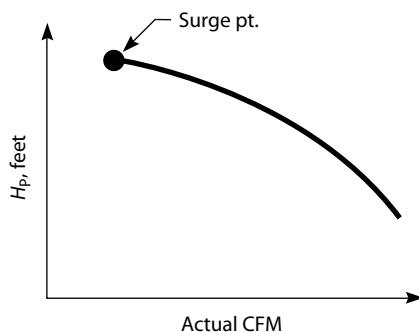


Figure 41.3 is similar to a centrifugal pump performance curve in regard to its shape. That is, it has a relatively flat portion of its curve and a relatively steep portion of its curve:

- Flat portion—That's the front portion of the curve. The important characteristic of the flat part of the curve is that a large reduction in flow produces a small increase in head. This means that an operating change that forces the compressor back up its curve will reduce the driver horsepower. Later in this chapter, you will see how critical this bit of information is. Most constant speed centrifugal compressors are designed to operate on the flat portion of their curves.
- Steep portion—That's the back portion of the curve, where a small reduction in flow produces a large increase in head. This means that an operating change that forces the compressor back up its curve will increase the driver horsepower.

When I describe how to save compressor horsepower by suction throttling later in this chapter, you will understand the importance of the shape of the performance curve. The concept of the suction throttle valve, in fixed-speed centrifugal compressors, is the least understood aspect of all process equipment.

41.1.3 Aerodynamic Stall

In my younger days, I used to try to meet women on airplanes. Finding myself seated next to an interesting lady, I would ask, "Have you ever wondered what makes this plane fly?" With this opening gambit, I would then explain:

This sketch (Fig. 41.4) is a cross-section of the wing. Because of the shape of the wing, the air has to travel a longer distance across the top of the wing than underneath the wing. This means that the velocity of the air as it travels across the top of the wing is greater than the velocity of the air as it travels underneath the wing. The energy to increase the velocity, or kinetic energy of the air as it flows across the top of the wing, does not come from the plane's engine. This energy

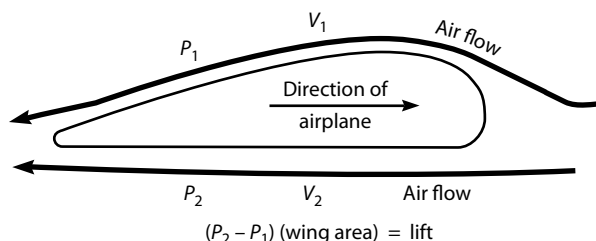


FIGURE 41.4 Aerodynamic lift.

to accelerate the air comes from the air itself; that is, the increase in the kinetic energy of the air flowing across the top of the wing comes from the barometric pressure of the air.

It follows, then, that the pressure on top of the wing (shown in Fig. 41.4) is less than the pressure underneath the wing. This difference in pressure, multiplied by area of the wing, is called *lift*.

As the plane's air speed is reduced, its ability to maintain a lift equal to its weight is reduced. At some reduced speed, the plane's lift then becomes insufficient to keep it flying. The aircraft undergoes aerodynamic stall. The plane falls out of the sky, crashes, and all the passengers are killed.

At this point, the young lady whom I was trying to impress would typically pick up a magazine and ignore me for the rest of the journey.

Surge is quite similar to aerodynamic stall. Of course, when a compressor surges, its rotor does not stop spinning. The rotor is spun by the motor. But when the flow of gas through the rotor falls below a certain rate, the forward velocity of the gas stops. With no flow, there is no velocity to convert to feet of head. Then the ΔP developed by the compressor falls to zero.

The discharge pressure of the compressor shown in Fig. 41.1 is 100 psig and its suction pressure is 10 psig. The gas flow, when the compressor surges, travels backward. The reverse gas flow pushes the rotor backward and slams it up against its thrust bearing. The suction pressure of the compressor increases and its discharge pressure decreases. Temporarily, the ΔP required to push the gas from the wet-gas drum and into the absorber shown in Fig. 41.1 is reduced. The polytropic head requirement is thus also temporarily reduced. The compressor may then run out on its performance curve, as it moves a greater ACFM volume, and moves away from surge. But in so doing, the compressor lowers its own suction pressure, raises its own discharge pressure, and creates the conditions for the next destructive surge.

41.1.4 Required ΔP

Movement of the gas from the wet-gas drum into the absorber requires a certain ΔP . According to Eq. (41.1)

$$\Delta P \approx (\text{vapor density}) \times (\text{polytropic head})$$

We can increase ΔP by either of the following options: raise the density of the vapor or raise the feet of polytropic head, developed by the compressor.

To raise the density of the gas, we could

- Raise the compressor's suction pressure
- Increase the molecular weight of the gas
- Decrease the temperature of the gas

We cannot change the density of the gas by altering the mechanical characteristics of the compressor.

To raise H_p , the feet of polytropic head, we could

- Increase the number of wheels on the rotor shown in Fig. 41.2
- Increase the diameter of the wheels
- Increase the speed of the rotor

We cannot change the feet of head developed by the compressor by altering the physical properties of the gas compressed.

Vapor density and feet of head are not related. But if the product of the two numbers does not result in sufficient ΔP to push the gas from the drum into the absorber, then the gas flow will stop. It will stop and then reverse its direction of flow. And that is surge.

41.1.5 Too Much Polytropic Head

You might conclude from my description of surge that the engineer needs to be cautious when designing a new compressor so that it will not surge. For example, let's assume that Jane has to issue the specifications for a new wet-gas centrifugal compressor. She checks with John, the unit engineer, for the proper molecular weight of the gas. John tells Jane that the molecular weight of the gas is normally 30, but it can be as low as 24—that is, the density of the gas can, on occasion, be 20 percent lower than normal.

Jane concludes that the lower-density gas will require more feet of polytropic head to develop the required ΔP . To avoid the possibility of surge, she decides to increase the number of wheels on the compressor from five to six. While Jane has used good engineering judgment, she has made a serious error. It turns out that John should not have been trusted. The actual molecular weight of the gas turns out not to be 24 or 30, but 36. The gas is 50 percent more dense than Jane's design specifications.

Poor Jane! The compressor's motor driver now trips off on high amps! In her efforts to avoid surge, she has run afoul of the real-world fact, that the motor amps required to drive a centrifugal compressor are approximately proportional to the molecular weight of the gas—in apparent contradiction to the Second Law of Thermodynamics.

I hope that you can now see the intimate relationship between surge in a centrifugal compressor and the amperage load on the motor used to drive the compressor. But let's see if I can prove that the Second Law of Thermodynamics is in harmony with our practical experience.

41.1.6 Effect of Molecular Weight on ΔP

Let us refer again to Fig. 41.1. Suddenly, there is an increase in the molecular weight of the wet gas. This causes the density of the gas to increase. This results in an increase of the compressor ΔP . As the compressor ΔP increases, the compressor's suction pressure decreases. Why? If the discharge pressure is kept constant by the absorber back-pressure control valve, then a bigger ΔP must drag down the suction pressure. The reduced suction pressure increases the suction volume (ACFM) of gas

flowing to the compressor. Why? Because a lower-pressure gas occupies a larger volume.

As the ACFM increases, we run out to the right on the compressor curve, shown in Fig. 41.3. As we move away from the surge point, the polytropic feet of head decreases. As the polytropic feet of head is reduced, the compressor ΔP comes partially back down to its initial value, until a new equilibrium is established. But because the initial disturbance of the equilibrium—the increased molecular weight—moved us away from surge, the new equilibrium will be established farther away from surge than the initial equilibrium. Not only will the new equilibrium be established farther away from surge, but the pressure in the wet-gas drum will wind up lower than the initial pressure in the drum.

Let's now assume that there is a sudden decrease in the molecular weight of the wet gas. This results in a decrease in the gas density. The ΔP developed by the compressor goes down. As a consequence, the compressor's suction pressure rises. This reduces the ACFM volume of gas flowing into the compressor. As the ACFM decreases, we back up on the compressor curve toward the surge point. As we move closer to surge, the polytropic feet of head developed increases. The compressor ΔP comes partly back up to its initial value, until a new equilibrium is established. But because the initial disturbance—the decreased molecular weight—moved us toward surge, the new equilibrium will be established closer to surge than the initial equilibrium. Also, the pressure in the wet-gas drum will wind up higher than the initial equilibrium pressure in that drum.

41.2 Compressor Efficiency

41.2.1 Maintaining a Constant Suction Drum Pressure

In the two examples we have just discussed, P_1 , the compressor suction pressure, was not being held constant. In fact, when the molecular weight of the gas increased, P_1 , the suction pressure, went down. Let me now rewrite Robert Mayer's equation, which I presented at the start of this chapter:

$$\text{Amps} \approx N \left[\left(\frac{P_2}{P_1} \right)^{0.23} - 1 \right] \quad (41.2)$$

This simplified version assumes that K (the C_p/C_v ratio) is a constant of 1.3; that T_1 , the suction temperature, is constant; and that the compression work is proportional to the motor amps (N is the number of moles of gas flowing into the compressor). (The K value of air and hydrogen at room temperature is 1.4. For CO_2 , K is 1.3. For ethane and heavier hydrocarbons, K gets progressively smaller, as the molecular weight increases. Methane is 1.31. Ethane is 1.22. K gets bigger as

temperature increases. Meaning the hotter and lighter the gas, the more difficult it is to compress.) From Eq. (41.2), note that a reduction in the suction pressure, P_1 , will cause the motor's amp load to increase.

But I said at the start of this chapter that I assumed that P_1 was to remain constant. Well, I made an impossible assumption. Because of the dynamic nature of the centrifugal compressor, it is simply impossible to permit the molecular weight of a gas to vary, and then state that none of the following parameters may change:

- Suction pressure, P_1
- Discharge pressure, P_2
- Number of moles, N
- Speed of the compressor

As the molecular weight goes up, P_1 goes down, the compression ratio (P_2/P_1) increases, and so does the work needed to drive the compressor. And the apparent contradiction between Dr. Mayer's equation and our plant experience is resolved.

But suppose we must maintain a constant pressure in the wet-gas drum. The pressure in this drum may be controlling the pressure in an upstream distillation column. To hold a constant pressure in the drum, we will have to resort to spillback suction pressure control, illustrated in Fig. 41.5.

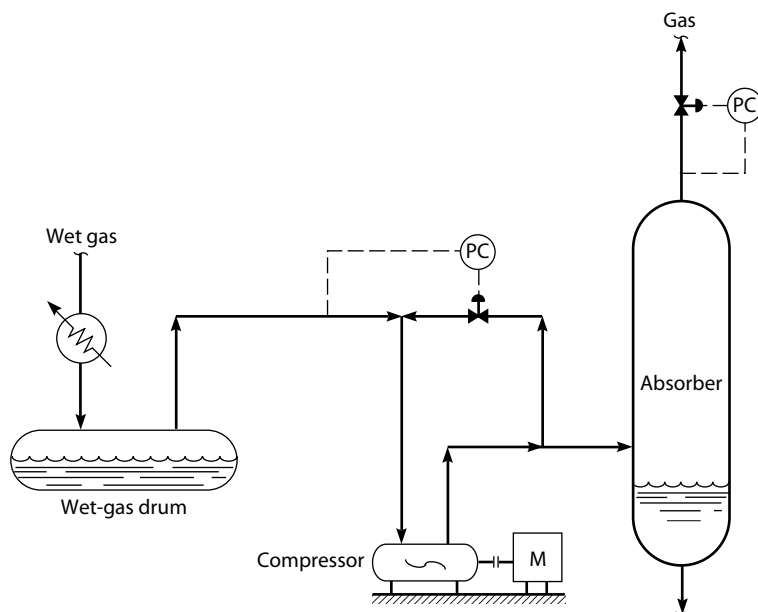


FIGURE 41.5 Spillback suction pressure control.

Again, let's assume that there is a sudden increase in the molecular weight of the gas. Again, the gas density increases. The ΔP developed by the compressor rises. The pressure in the wet-gas drum drops. The spillback pressure-control (PC) valve starts to open. The number of moles, N , flowing to the compressor increases. The ACFM also increases. The compressor now runs out on its operating curve (Fig. 41.3) to the right, away from surge. The polytropic feet of head H_p drops. The ΔP is brought back down to its initial value. This keeps both the compressor suction pressure and the pressure in the wet-gas drum constant.

Great. But what about Dr. Mayer? What about Eq. (41.2)? It is true that the compression ratio (P_2/P_1) has been held constant. But as the molecular weight has increased, the number of moles N has also increased. Why? Because, the spillback valve has opened. And as N increases, so does the amperage drawn by the motor.

Why, then, in ordinary process plant practice, do we see an increase in the amps on a motor driving a centrifugal compressor as the gas becomes heavier? Does it take more work to compress a mole of propane (44 MW [molecular weight]) than it does to compress a mole of methane (16 MW)? Certainly not. It's just that compressing a heavier gas forces the spillback to open to prevent the pressure from falling in the wet-gas drum. This extra gas recirculating through the compressor is the real factor that increases the amp load on the motor driver.

41.2.2 Variable-Speed Driver

Wouldn't it be simpler to just slow down the compressor as the molecular weight increases? After all, it seems as if we need less feet of head when the gas density increases to maintain a given ΔP . As we slow the compressor by 10 percent, the feet of polytropic head would drop by 20 percent. Fine. But 99 percent of motor-driven compressors are fixed-speed machines.

Does this mean that we would be better off driving a large centrifugal compressor with a variable-speed driver? Perhaps with a steam turbine or gas-fired turbine. You bet! Especially when the molecular weight is highly variable.

41.2.3 What about Jane?

I hope you all understand the mistake Jane made. She trusted John. But if she had it all to do over again, what could Jane have done differently? She has to drive the compressor by a fixed-speed motor. The gas molecular weight is going to be unpredictable. The pressure in the wet-gas drum has to be kept constant.

Figure 41.6 is a partial answer to the problem of excessive power demand on the compressor's motor driver. We use the suction throttle valve (i.e., the "PC," control valve) shown in this figure. The use of the compressor suction throttle valve will partly offset the increase in the amp load on the compressor's motor driver, which results from

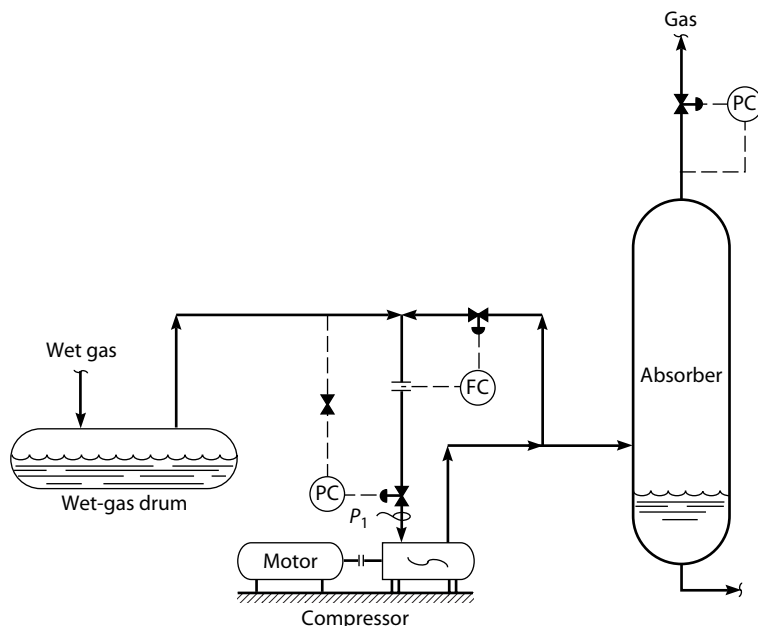


FIGURE 41.6 Suction throttling pressure control.

an increase in the molecular weight of the wet-gas flow to the compressor. As I will explain below, the beneficial effects of suction throttling are a function of the shape of the compressor's performance curve. As long as the compressor is operating on the flat portion of its curve (see Fig. 41.3), suction throttling helps. If the compressor is operating on the steep portion of its curve (which is not normal), suction throttling will not help to reduce the amperage load on the driver.

If the molecular weight of the gas increases, the gas density will increase. The ΔP will increase. The pressure in the wet-gas drum will drop. The new suction throttle PC valve will start to close. This will restore the pressure in the wet-gas drum without increasing the flow of gas through the spillback valve. But what happens to P_1 ? How does closing the suction throttle valve affect the actual compressor suction pressure?

Gentle reader, take a break. You are up to the hardest part of this book. Have a Coke, and we will continue.

41.2.4 Damming the River Yeo

The River Yeo is a gentle stream that meanders through western England. In the seventeenth century, the river was used to move coal-laden barges. But near the town of Yeovil, the river became too shallow for deep draft barges to negotiate. So a dam was proposed.

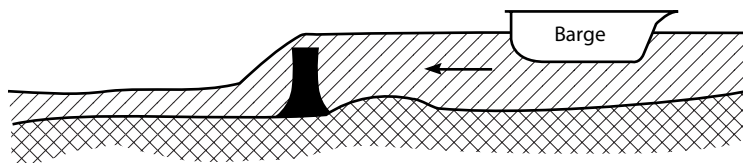


FIGURE 41.7 Dam on a river does not affect level downstream.

This would raise the height of water upstream of the dam, as shown in Fig. 41.7.

The farmers downstream of the dam were concerned that the river flowing through their fields would run dry. The engineer who designed the dam assured them that this would not happen. Certainly, at first the flow of water downstream of the dam would be just a trickle. But after a few months, when a new equilibrium was established, water would begin overflowing the dam at the same rate that water formerly flowed down river. The level of the river downstream of the dam would then be the same as it had been before the dam existed.

The suction throttle valve, shown in Fig. 41.6, is analogous to the dam on the River Yeo. When we close this valve, the pressure in the wet-gas drum will increase. But the pressure at P_1 , the compressor suction pressure, at equilibrium is completely unaffected by the movement of the suction throttle valve.

For example, let's say that the molecular weight of the gas increases by 20 percent. The ΔP developed by the compressor would then increase by 20 percent, from 90 to 108 psi. If the absorber pressure is fixed, then the wet-gas drum pressure would drop by 18 psi. To prevent this excessive decrease in the drum pressure, we throttle on the suction PC control valve. The pressure drop through this valve would increase by 18 psi. But the pressure at P_1 will drop by 18 psi, just as if we never moved the suction throttle valve.

Controlling the pressure in the drum by suction throttling allows the compression ratio P_2/P_1 to increase in response to an increase in the molecular weight of the gas. As the compression ratio increases, according to Eq. (41.2), the amperage on the motor driver also increases. But not by a lot.

According to Eq. (41.2), the compression ratio is raised to a small fractional exponent (viz., 0.23). So even if the compression ratio goes up a lot, the amp load on the motor driver will increase by very little.

The main reason, however, that suction throttling reduces the amp load on the motor driver is due to the shape of the compressor performance curve (see Fig. 41.3). If we are operating on the flat portion of the curve, suction throttling will have the following effect:

- Flow will go down a lot.
- Delta P (or polytropic head) will go up just a little.

Meaning, suction throttling just a bit causes the spillback valve to throttle back a lot. Or, N , the number of moles of gas compressed, will drop a lot, but P_2/P_1 , the compression ratio, will increase just a little. As we normally operate on the flat portion of the compressor curve, suction throttling will reduce the amp load on the driver. If, however, we were operating on the steep portion of the compressor curve, suction throttling could increase the amp load on the motor driver.

To summarize, suction throttling moves us closer to surge, but saves energy by forcing the spillback valve to close.

Of course, we may also control the wet-gas drum pressure very nicely by the spillback PC valve, shown in Fig. 41.5. But this mode of control causes N , the number of moles in Eq. (41.2), to rapidly increase. And the motor amps will increase about as fast as the number of moles.

The useful rule of thumb is then

- When the molecular weight rises, the motor amps must increase.
- If the molecular weight rises by 20 percent and we control the drum pressure by spillback control, the motor amps will increase by about 20 percent.
- If the molecular weight rises by 20 percent and we control the drum pressure by suction throttle control, the motor amps will increase by about 10 percent.

41.2.4.1 Stone walling

Surge is bad and will eventually damage a compressor's rotor and case internals. Stone walling sounds equally bad, but it is not. Operating in a stone walling mode simply means you have reduced the compressor's discharge pressure to such a great extent that a further reduction in discharge pressure will not result in any increase of gas flow. Anyone who has operated a centrifugal pump will recognize this condition. That is, past some pump discharge valve position, opening the valve further will not increase the flow. I like to say that you have fallen off the curve.

41.2.5 Saving Electricity

Forget about Jane. Forget about John. At the moment, let's assume that there is nothing wrong with the compressor shown in Fig. 41.6. We are operating far away from the surge point, and the motor is pulling only 70 percent of its maximum amp load.

Both the suction throttle pressure control (PC) valve and the spillback (flow-control [FC]) valve shown in Fig. 41.6 are in a nice operating position. All we wish to do is save electricity.

This is done by reducing the set point on the FC valve. As we do this

1. The FC valve partly closes.
2. The number of moles compressed N is reduced.

3. The volume of gas compressed (ACFM) goes down.
4. According to the compressor curve shown in Fig. 41.3, the feet of polytropic head H_p increases as the ACFM decreases.
5. The increased H_p increases the ΔP developed by the compressor.
6. This reduces the pressure in the wet-gas drum.
7. The PC valve shown in Fig. 41.6 now begins to close, to restore the pressure in the wet-gas drum.

The net result of these changes is to reduce N and increase P_2/P_1 . Because of the shape of the compressor performance curve, this should result in a reduction in the amp load of the motor driver. A rough rule of thumb is that for every 10 percent decrease in N , the total number of moles compressed, the amp load on the motor driver will fall by 5 percent.

But as we reduce N , we are moving toward the surge point in Fig. 41.3. If we try to save too much electricity by forcing the FC to close too much, then the centrifugal compressor may be forced into surge. That is why we call the FC spillback valve the *antisurge valve*.

Notice how the problem of surge and the amp load on the motor driver are two ideas that interact together. It is kind of like solving two equations with two unknowns. The two objectives we are trying to optimize are the motor amps and avoidance of surge. The two handles we have on the problem are the suction throttle valve and the spillback valve. And just like solving the two equations, we have to optimize the position of the two valves simultaneously.

41.2.6 Suction Temperature

Throughout our discussion of motor-driven, constant-speed centrifugal compressors, I have assumed that the suction temperature was constant. But let's refer back to Fig. 41.1. Assume that the temperature in the drum increases. How will this affect the pressure in the drum?

If the increased temperature T_1 increases the density of the gas, then the ΔP developed by the compressor will go up, and the drum pressure will decline. Let's assume that the feed to the drum is a mixture of

- Ethane
- Propane
- Butane
- Pentane
- Gasoline

As the drum temperature increases, the heavier components in the liquid phase are vaporized into the vapor phase. This increases the molecular weight of the vapors and hence the vapor density.

Now, let's assume that the feed to the drum is a mixture of hydrogen and heavy mineral oil.

As the drum temperature increases, the molecular weight of the gas remains at about 2, the molecular weight of hydrogen. The mineral oil is too heavy to vaporize. The density of the vapor goes down, because the vapor is hotter. The drum pressure then increases.

41.2.7 Sulfur Plant Air Blower

Here is a problem that came up on a sulfur recovery facility in Punto Fijo, Venezuela. The combustion air blower, shown in Fig. 41.8, was a fixed-speed, motor-driven centrifugal machine. The air intake filters were severely fouled. They had a pressure drop of about 8 inches of H_2O . The atmospheric vent valve, used to control the discharge pressure at a constant 12 psig, was 50 percent open. The unit engineer had been asked to calculate the incentive in electrical power savings for cleaning the filters.

Cleaning the filters would have raised the suction pressure to the blower. This would have reduced the blower's compression ratio. But because the blower is a fixed-speed centrifugal compressor, the amount of air compressed N would have increased. The atmospheric vent discharge valve would have been forced open. The greater volume of air compressed would have caused the compressor to work harder, even though the compression ratio (P_2/P_1) was lower. In conclusion, cleaning the filters would have increased the amps drawn by the motor.

To prove my point, I slid a piece of plywood across one filter. The PC vent valve, shown in Fig. 41.8, began to close, and the amps drawn by the motor driver went down!

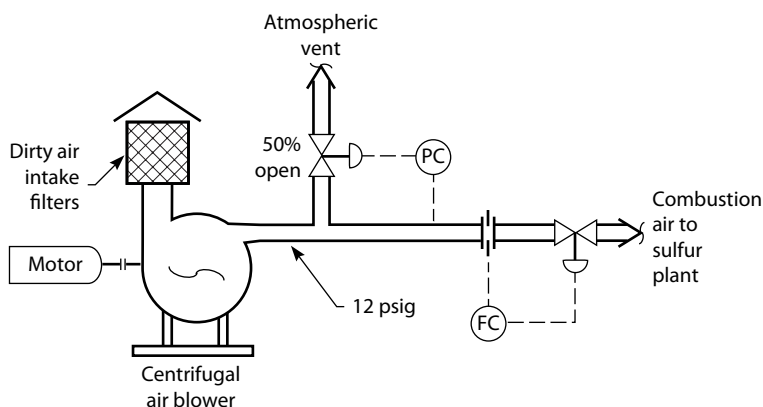


FIGURE 41.8 Filter plugging affects motor amps.

41.2.8 Centrifugal Compressor Check Valves and Surge Protection

A motor-driven centrifugal compressor may be forced into surge due to a combination of several factors:

- Excessive suction temperature
- Low molecular weight
- Excessive discharge pressure
- Low flow
- Suction pressure too low
- Low polytropic head due to defective or fouled rotor wheels

When the compressor surges, gas flow stops. However, as the compressor discharge pressure will still be high, the gas flows backward through the wheels of the compressor case. The momentum of this reverse gas flow pushes the rotor backward. The check valve on the compressor discharge greatly retards the force of this reverse gas flow.

On my sulfuric acid alkylation unit refrigeration compressor, which I operated from 1974 through 1976 in Texas City, I found the horrible banging sound of this check valve to be the most worrying aspect of a surge. Each violent bang frightened me. However, what I failed to recognize at the time was that it was the violent closing of the check valve that protected my alkylation unit refrigeration compressor from destruction due to the surging.

In general I have found that compressors operate at:

- Low speed, less than 3600 rpm (three-phase motor speed)
- Low overall compression ratios (less than 2 or 3 to 1)
- Reasonably high molecular weights (30 to 40 or higher)

Such machines can operate for many minutes (but not hours) in a surging mode without sustaining noticeable mechanical damage to the rotor, if there is a check valve on the compressor's discharge.

However, keep the following potential problem in mind. This incident occurred at a plant in South Africa on a catalyst regeneration recycle gas compressor.

- The operators closed the discharge to the suction recycle valve, which reduced the flow and raised the discharge pressure.
- The machine started to surge.
- The reversal of gas flow raised the suction temperature, which caused the motor driver to automatically trip-off on high temperature for surge protection.

- The primary lube oil pump for the bearings was driven by the shaft of the recycle gas compressor. So it shut-down with the compressor.
- But the backup lube oil pump, which was supposed to start automatically on low lube oil pressure to the bearings, failed to start up on auto.
- With a loss of lube oil pressure to the bearings, the bearings were rapidly damaged, and the resulting vibrations destroyed the gas compressor.

The plant supervisor claimed that the destruction was due to surge. This was not true. The destruction was due to the supervisor failing to verify that his backup lube oil pump would start up on auto, on low lube oil pressure to the bearings. (See *Troubleshooting Process Plant Control*, Wiley, 2009 for details as to how this is done.)

41.3 Frequently Asked Questions about Centrifugal Compressors

Many of the questions I receive from students pertain to centrifugal compressor operations. I've summarized the more frequently asked questions and answers below (refer to Fig. 41.9):

Q: What causes surge?

A: Due to low gas molecular weight, or too high a ΔP required, or too high gas temperature, the compressor head requirement gets too high for the gas to be pushed through the compressor case by the spinning rotor. The gas flow then stops and reverses.

Q: What's stone walling?

A: Reducing the discharge pressure does not increase flow.

Q: Is stone walling damaging to compressors?

A: No.

Q: Is surge damaging to compressors?

A: Yes. Each surge deforms the thrust bearing.

Q: Does suction throttling save energy?

A: If you are operating (see Fig. 41.9) in the normal range of the compressor, that is, on the flatter portion of the performance curve, suction throttling will save energy because flow will be reduced faster than head goes up. But, if you are operating on the steep part of the curve, suction throttling can waste energy because head goes up faster than flow goes down. The real effect of suction throttling can only be predicted by looking at the compressor performance curve.

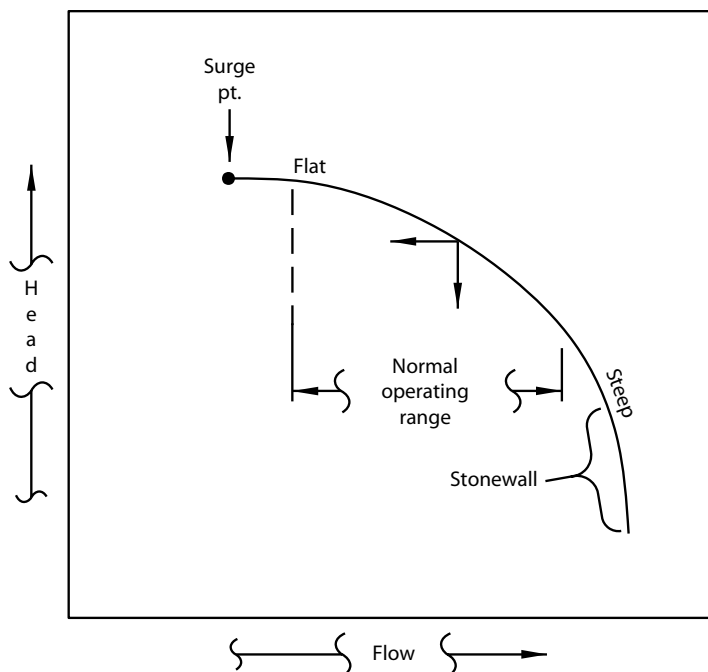


FIGURE 41.9 Centrifugal compressor curve showing effect of stonewalling.

Q: Opening up the spillback on my compressor has caused it to surge. I thought that should prevent surging?

A: You're spilling back hot gas from the compressor discharge. This lowers the gas density. Any factor that lowers gas density will promote surging.

Q: If a turbine driven compressor starts to surge, should I make it go faster or slower?

A: There's no general answer. You need to look at your compressor surge vs. speed curves. See what happens as you change speed as to whether or not you are closer to the surge point. On my sulfur plant air blower in Texas City, depending on operating conditions, I could cause the blower to surge by increasing or decreasing speed.

(*Caution:* The equations presented in this chapter for polytropic head and compression work have been simplified for clarity. They cannot be used for rigorous engineering calculations.)

CHAPTER 42

Reciprocating Compressors

The Carnot Cycle; Use of Indicator Card

Centrifugal compressors are dynamic machines. They convert velocity, imparted to the gas by a spinning wheel, to polytropic feet of head. The feet of head multiplied by the density of the gas equals the pressure boost produced by the centrifugal compressor.

A *reciprocating compressor* is a direct volume-reduction machine. The gas is simply squeezed out of a cylinder by a piston and pushed into the discharge line. The molecular weight of the gas does not influence the suction or discharge pressure of the compressor. The gas density does not influence the compressor performance or the work required by the driver.

The reciprocating compressor is a positive-displacement compressor. It is cheaper to purchase and install than a centrifugal compressor. It is also—in theory—far more efficient (90 percent) than a centrifugal compressor (70 percent). Certainly, reciprocating compressors are more simple to understand and engineer than centrifugal machines. Best of all, they are not subject to surge.

There are only two real problems with reciprocating compressors: pulsation and mechanical reliability. But these problems are so intractable that, for most industrial applications, centrifugal compressors are preferred; the exception is when dealing with low molecular weight gas. A low molecular weight gas, such as hydrogen, has a low density. Let's say that a compressor must develop a large differential pressure or ΔP . Recall Eq. (41.1) from the previous chapter:

$$\Delta P = (\text{gas density}) \times H_p$$

where H_p = polytropic feet of head.

A centrifugal compressor developing a lot of ΔP with a low gas density needs to produce a lot of polytropic feet of head. This means that the centrifugal compressor must

- Be a high-speed machine
- Have a lot of wheels or stages
- Have large-diameter wheels

The most cost-effective way to produce a high polytropic head is to increase the number of wheels on the rotor. But the longer the rotating assembly, the more difficult it is to properly balance the rotor. Especially for high-speed machines, rotors that become unbalanced are subject to destructive vibration. Therefore, for low molecular weight gas services (less than 10), it is common to use reciprocating compressors.

Reciprocating compressors are favored over centrifugal machines for temporary installations such as gas field well-head service. Also, when low initial capital investment is favored over long-term maintenance costs, reciprocating compressors are often used.

42.1 Theory of Reciprocating Compressor Operation

Figure 42.1 is a simplified sketch of a cylinder of a reciprocating compressor. The cylinder is shown as a single-acting cylinder. Typically cylinders are *double-acting*, meaning that there are valves on both ends of the cylinder and that the piston is compressing gas, in turn, on both ends of the cylinder.

The far end of the cylinder is called the *head end*. The end of the cylinder nearest the central shaft is called the *crank end*. I have shown the valves only on the head end to simplify my description of the compressor's operation.

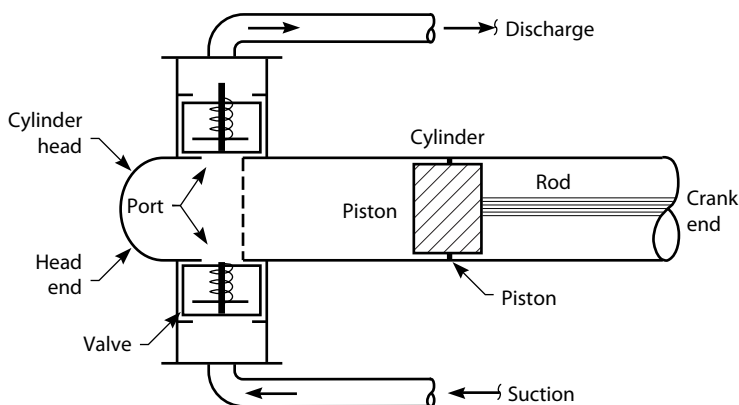


FIGURE 42.1 Reciprocating compressor cylinder.

There are four distinct steps in the compression cycle of the cylinder, shown in Fig. 42.1: compression, discharge, expansion, and intake.

42.1.1 Compression

The initial position of the piston is as far from the cylinder head as possible. This is the start of the compression stroke. This position is called *top dead center*. The piston now moves toward the cylinder head. The pressure of the gas inside the cylinder increases.

42.1.2 Discharge

At some point, as the piston approaches the cylinder head, the gas pressure inside the cylinder exceeds the pressure in the discharge line. The discharge valve now springs open, and gas is pushed out of the cylinder into the discharge line.

Both the discharge valve and the intake or suction valve are simply spring-loaded check valves. The discharge valve only permits gas to be forced out of the cylinder. The intake valve only permits gas to enter the cylinder.

The piston continues its travel toward the cylinder head. At some point it stops and reverses its direction. This point is called *bottom dead center*, indicated by the dotted line in Fig. 42.1. Of course, bottom dead center cannot coincide with the end of the cylinder. The piston would have to travel past the valve ports for this to occur. If the piston travels past the discharge port, the compressed gas could not be pushed out of the cylinder into the discharge line. So bottom dead center must line up near the crank-end edge of the valve ports, as shown in Fig. 42.1.

This means that a substantial volume of gas remains trapped between the cylinder head and the piston before the travel of the piston is reversed. This volume of gas is called the *starting volumetric clearance*. It determines the volumetric efficiency of the reciprocating compressor. We will discuss this later.

42.1.3 Expansion

As soon as the piston reverses its direction of travel, the pressure of the gas inside the cylinder drops. The gas pressure drops below the discharge line pressure, and the spring-loaded discharge valve slams shut. The piston continues its travel toward the crank end of the cylinder. The pressure of the gas inside the cylinder continues to fall, but the suction or intake valve remains closed.

42.1.4 Intake

At some point, as the piston approaches the crank end, the gas pressure inside the cylinder falls below the pressure in the suction line. The suction, or intake, valve now springs open, and gas is drawn out of the suction line and into the cylinder. This portion of the intake stroke continues until the piston returns to top dead center.

42.2 The Carnot Cycle

The piston has now completed its cycle. I have drawn a picture of this cycle in Fig. 42.2. I have plotted the pressure inside the cylinder against the volume of the gas inside the cylinder. The “gas inside the cylinder” refers to the gas between the piston and the cylinder head.

Beginning at top dead center:

1. The gas is compressed up to the discharge pressure.
2. The gas is pushed out into the discharge line at a pressure equal to the discharge-line pressure.
3. The piston reaches bottom dead center and reverses its direction.
4. The gas left in the end of the cylinder expands—and depressurizes—until the suction-line pressure is reached.
5. New gas is taken into the cylinder until the piston returns to top dead center.

At any point in this cycle, there is a pressure inside the cylinder that corresponds to the volume of gas inside the cylinder. That is all the Carnot cycle chart, drawn in Fig. 42.2, means.

Professor Nicolas L. S. Carnot, in the late nineteenth century, realized that the area inside the plot of pressure vs. volume represented the work needed to compress gas in a reciprocating compressor. In other words, the change of pressure multiplied by the change in volume is equal to the work done by the piston on the gas. Professor Carnot

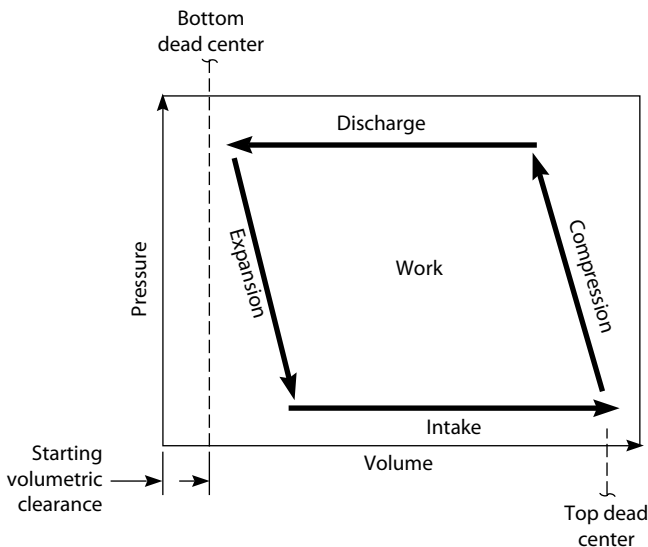


FIGURE 42.2 The Carnot cycle.

called this *PV* (pressure vs. volume) work. He then used calculus to sum up the area inside the lines shown in Fig. 42.2. The total area is now called *ideal compression work*.

What's wrong with Carnot's theory? I have nothing against Professor Carnot personally. I do not hold his French nationality against him. I do not dislike French people, even though Liz's purse was stolen in Paris. But I do have several problems with the Carnot cycle:

- Valve leakage
- Spring tension
- Pulsation

The Carnot cycle plot represents ideal compression work. But we in the process industry have to worry about actual compression work and the loss of compression efficiency caused by these three problems.

We would like to understand the actual compressor cycle, rather than the idealized Carnot cycle. We would like to see an actual plot of the pressure inside the cylinder, compared to the volume inside the cylinder. And there is a way to do this.

42.3 The Indicator Card

The pressure inside the cylinder can be measured with a *pressure transducer*, a device that converts a pressure into an electrical output. The pressure transducer screws into the end of the cylinder head. There is a screwed plug in the cylinder head for this purpose.

The volume of gas inside the cylinder is a function of the piston position. The piston position is a function of the crank shaft position. The crank shaft position can be measured by a magnetic pickup attached to the crank shaft. Anyone who has had an automobile spark plug firing timing adjusted electronically is familiar with this method of determining piston position.

The output from the magnetic pickup and the output from the pressure transducer are connected to an oscilloscope. After a computer conversion of the data, the resulting pressure/volume plot is printed out. This plot is called an *indicator card*. The indicator card shown in Fig. 42.3 was generated from a 4000-hp natural-gas compressor near the town of Hebronville, Texas.

The solid line is the indicator card plot. The dotted line is the Carnot or ideal compression work cycle, which I have drawn myself. The piston position, shown on the horizontal axis, is proportional to the volume of gas inside the cylinder.

The area enclosed by the solid line is the total or actual compression work. The area enclosed by the dotted line is ideal or useful compression work. The area between the dotted line and the solid line represents compression work lost to heat. The area inside the dotted line divided by the area inside the solid line is called *adiabatic compressor efficiency*.

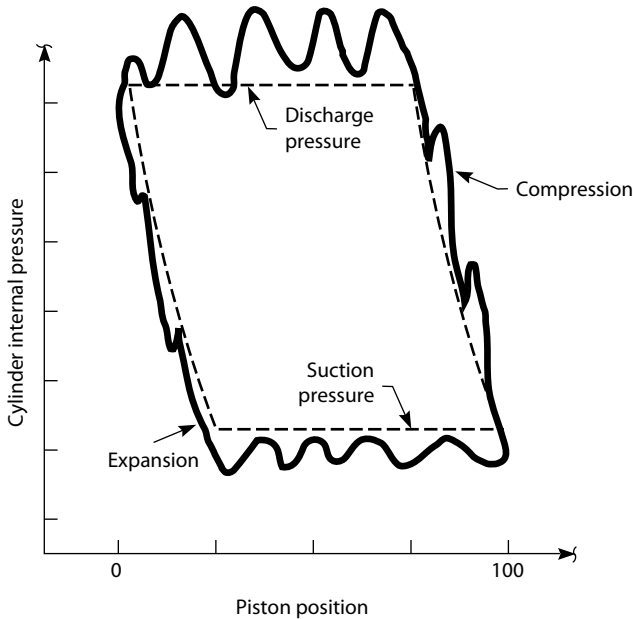


FIGURE 42.3 An indicator card plot.

42.3.1 Pulsation Losses

Take a closer look at the discharge portion of the indicator card shown in Fig. 42.3. Note that the cylinder internal pressure rises well above the discharge-line pressure before gas is pushed out of the cylinder and into the discharge line. This is a consequence of the spring tension of the discharge valves.

In order to force these valves open, a certain amount of extra pressure in the cylinder is needed. The extra pressure overcomes the spring tension. Once the valve is open, gas flows out of the cylinder into the discharge line, and the cylinder pressure falls. But, as shown on the indicator card, the discharge valve apparently opens and closes five times during one cycle.

This multiple opening of the discharge valve is called *valve pulsation* or “valve flutter.” The peaks on the indicator card caused by this pulsation are a source of considerable energy waste. The amplitude (i.e., the height of the peaks) is a function of the valve spring tension. The frequency of the peaks (i.e., the number of peaks) is a function of the speed of the compressor and the geometry of the suction and discharge piping.

We can reduce the amplitude (height) of the peaks by using weaker springs in the discharge valve. I tried this at the Hebronville compressor station, and it did work. The frequency (number) of peaks cannot be altered by any cost-effective method.

When we reduce the spring tension, we prevent the valves from closing as tightly as they did with stronger springs. The discharge valves are now more prone to leakage. The effect of discharge-valve leakage can be seen in the expansion portion of the cycle, shown in Fig. 42.3.

The sudden jumps in pressure are caused by high pressure gas blowing back from the discharge line into the cylinder. If the discharge-valve leakage gets bad enough, the amount of gas compressed may approach zero. While the amount of actual or total compression work may still be quite high, the amount of ideal or useful compression work may be essentially zero.

42.3.2 Using the Indicator Card

Is it then a good idea to make the discharge-valve springs weaker? Weaker springs cut down on pulsation losses, but also increase valve leakage losses. If we consult the indicator-card plot, we can see that the pulsation losses (for the discharge portion of the cycle) greatly exceed the valve leakage losses (for the expansion portion of the cycle). It therefore seems that weaker springs probably are a good bet to promote greater compressor efficiency.

I could repeat the same story for the intake or suction valves. Leakage of the intake valves would appear as peaks on the compression portion of the cycle. Pulsation losses for the intake valves would appear as peaks on the indicator card during the suction or intake portion of the cycle.

Other problems that can be identified by use of the indicator card are

- Piston ring leakage
- Excessive valve velocity losses
- Breakup of the valve plates

The indicator card is the only real way to monitor reciprocating-compressor performance. Typically, the equipment and personnel to generate the card can be obtained from a local company specializing in this service. Often, the indicator card is referred to as a *beta scan plot*.

42.4 Volumetric Compressor Efficiency

Reciprocating compressors have two sorts of compressor efficiency:

- Adiabatic
- Volumetric

So far, we have limited our discussion to adiabatic compression efficiency. This inefficiency downgrades work to heat. For a given compression ratio, the temperature rise of the gas as it flows through the compressor may be excessive, thus indicating a low adiabatic compression efficiency. Both centrifugal and reciprocating compressors suffer from this common problem, which is the subject of Chap. 43.

Volumetric efficiency applies only to reciprocating compressors. A reduction in volumetric efficiency reduces the gas flow through the compressor. A reduction in volumetric efficiency need not reduce the adiabatic compression efficiency.

A reduction in volumetric efficiency reduces the work required from the driver. For reciprocating compressors, we intentionally reduce the volumetric efficiency to reduce the load on the driver. Sometimes this is done to save energy; sometimes it is done to prevent the motor driver from tripping off on high amps.

42.5 Unloaders

In the plant, we use the term *unloading* to indicate various ways of reducing the volumetric efficiency of a reciprocating compressor. There are two sorts of unloaders:

- Valve disablers
- Head-end clearance pocket adjusters

The proper way to reduce the volumetric efficiency is to increase the starting volumetric clearance. This is done with an adjustable unloading pocket, as shown in Fig. 42.4. This device, also called the *head-end unloader*, works by increasing the starting volumetric clearance. The starting volumetric clearance is the volume of gas trapped between the cylinder head and the piston when the piston position is at bottom dead center (see Fig. 42.2).

Turning the wheel at the back end of the cylinder counterclockwise pulls back a large internal plug in the head. Now, when the piston starts to withdraw toward the crank end of the cylinder, there is more gas left inside the cylinder to expand. The greater the volume of gas inside the cylinder when the piston is at bottom dead center, the closer the piston is to top dead center before the intake valve opens.

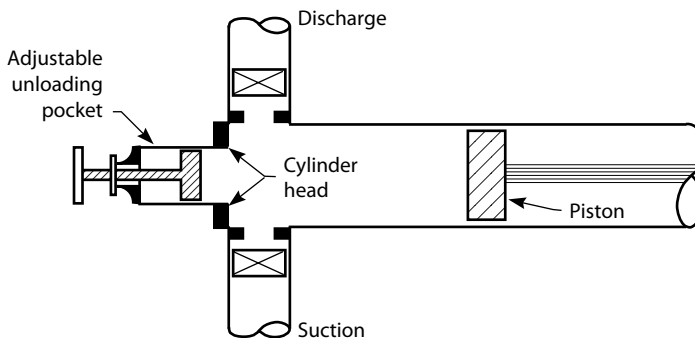


FIGURE 42.4 An unloading pocket reduces engine load and volumetric capacity.

The delay in the opening of the intake valve reduces the amount of gas drawn into the cylinder. This reduces the number of moles of gas compressed by the piston. Compression work is also diminished, and the driver horsepower or motor amp load drops.

42.5.1 Valve Disablers

The head-end unloader has a great advantage in that it unloads the driver without increasing the amount of work required per mole of gas compressed. The disadvantages of the adjustable clearance pocket or head-end unloader, shown in Fig. 42.4, are

- It is an expensive added feature.
- It cannot be used on the crank end of the cylinder.

In practice, the head-end unloader can reduce compression work required per cylinder only by roughly 30 percent.

The second way to unload a reciprocating compressor is with valve disablers. Most of the unloaders you have on your compressors are likely of this inferior type. They typically consist of steel fingers, which are pressed down through the valve-cap assembly. The fingers prevent the valve plate from moving. These valve disablers are far less costly than an adjustable clearance pocket. They may be used on both the crank and head ends of the cylinder. They can reduce the gas flowing through a cylinder to zero.

Unfortunately, valve disablers have a detrimental effect on the adiabatic compressor efficiency. This means that, even though no gas may be moving through the crank end of a cylinder, the piston is still doing work on the gas inside the crank end of the cylinder. If you would like proof, place your hand on the valve cap on such a disabled cylinder. The high temperature you will feel is wasted compression work going to useless heat. I have measured in the field that, after a cylinder end is completely disabled, it is still converting 20 percent of the former compression work to heat.

The other problem with valve disablers is poor mechanical reliability. They get stuck inside the valve cap, especially in dirty-gas service. Once stuck, the valve—and really the entire end of the cylinder—is out of service until the next overhaul of the reciprocating compressor.

42.5.2 Valve Failure

The weak points of a reciprocating compressor are the intake and discharge valves. Valves fail because the movable valve plate or the springs break. The cause of failure is usually liquid entering the compressor. Reciprocating compressors do not like liquids. A small, well-dispersed liquid flow to a centrifugal compressor will help keep its rotor clean and does no damage. The same amount of liquid will tear up the valves on a reciprocating machine.

High temperature also hastens spring failure and plate cracking of the discharge valves. The primary cause of high discharge-valve temperature is valve leakage. Leaking valves cause part of the gas flow to be recompressed by the piston. This extra compression work can result in some very hot discharge valves.

Valve leakage is caused by a combination of pulsation and fouling deposits. Sulfur, sulfur salts, ammonium chloride, and other volatile compounds may all sublime inside the valve assemblies. The word *sublime* means to change directly from a vapor into a solid. Amine salts, polymers, oxygen, diolefins, water, and products of corrosion also contribute to valve fouling. These deposits accumulate between the moving valve plate and the fixed valve slotted face. They prevent proper seating of the movable valve plate. This results in valve leakage. Centrifugal compressors are far more tolerant of fouling than reciprocating compressors.

42.6 Rod Loading

The piston rod used in a reciprocating-compressor cylinder can fail if overstressed. The stress experienced by the piston rod is a direct function of the ΔP developed by the cylinder. That is because on one side of the piston, we have the suction pressure, and on the other side of the piston, we have the discharge pressure. The ΔP times the area of the piston is therefore the force acting on the piston. Excessive force will break the piston rod.

The high-discharge-temperature trip on your reciprocating compressor is intended to prevent the piston rod from being overstressed. The higher the ΔP , the greater the temperature rise of the compressed gas. The compressor manufacturer has calculated the expected temperature rise of the gas for the maximum ΔP to which the piston should be exposed. The high-temperature trip is not intended to protect mechanical components from excessive wear and tear, due to high temperature. It is there to protect the machine from the very serious consequences of piston-rod failure.

Do not reset this high-temperature trip above the manufacturer's specifications without prior consultation with the equipment manufacturer.

42.7 Variable Molecular Weight

My client had installed a reciprocating compressor to supply feed gas to a hydrogen production facility. The design gas composition was:

- 50 percent methane
- 50 percent ethane
- Molecular weight = 24

However, after placing the purchase order for the new compressor, the design basis was changed. The new design gas composition was:

- 82 percent hydrogen
- 10 percent methane
- 8 percent ethane
- Molecular weight = 6

My client asked me if they should cancel the order for the new machine and specify a revised compressor. I advised them to forget about the difference between the two cases. It would not make much difference to a reciprocating compressor.

In both cases, the volume of gas (or number of moles) compressed would be about the same. I calculated that due to a larger compressibility factor, the hydrogen-rich gas case would reduce the number of moles of gas compressed by about 2 percent, compared to the first methane-ethane mixture case.

Further, I calculated that due to a greater ratio of the specific heats:

$$C_p/C_v = K$$

the hydrogen-rich gas case would increase compression work per mole of gas by about 5 percent, compared to the methane-ethane mixture case. As the motor driver was oversized by 30 percent, the extra compression work was of little significance.

Based on my evaluation, the new reciprocating compressor was purchased as originally specified for the 24 molecular weight gas case. When the machine was put in service, with its design 50 percent methane-50 percent ethane gas feed, it handled its design rate of 400,000 SCF/hr (standard cubic feet per hour). However, the preferred hydrogen plant feed was the gas with 82 percent hydrogen (i.e., the 6 molecular weight gas). When the operators switched to the hydrogen-rich gas, the flow indication of feed gas to the hydrogen plant dropped to 200,000 SCF/hr.

The operators at the hydrogen plant immediately radioed the outside operator at the new feed gas reciprocating compressor that they were losing their feed gas flow. Quickly, the outside operator swung back to the 24 molecular weight gas stream, and flow was restored to 400,000 SCF/hr.

42.7.1 Effect of Gas Density on Flow Indication

Please do not think I'm making this story up. It's true. The location was the Coastal Refinery in Aruba. The engineer assigned to this project spent \$85,000 for 16 new compressor valves for the reciprocating compressor. These valves had a somewhat smaller valve plate lift to accommodate the lower molecular gas. Of course, this sort of minor fine-tuning made no noticeable difference. Switching from 24 to

6 molecular weight gas still caused the indicated flow to the hydrogen plant to drop by half.

The process manager convened a meeting to discuss this disaster. Inevitably, I was blamed for the poor efficiency of the compressor when handling the hydrogen-rich gas. In my defense, I presented the following calculation for gas flow measured by an orifice plate:

1. Flow is proportional to the square root of the pressure drop through the orifice plate.
2. Pressure drop through an orifice plate is proportional to gas density. That is, if the gas density doubles, so will the observed pressure drop.
3. Gas density is proportional to the molecular weight. If the molecular weight increases from 6 to 24, the density of the gas will increase by a factor of four.
4. Therefore, gas flow measured by an orifice plate is proportional to the square root of the molecular weight of the gas flowing through the orifice plate.

I then explained to the plant manager that the flow of gas through the orifice plate flowmeter had always been constant. The smaller indicated flow of hydrogen-rich gas was not real. My client had forgotten to adjust the flowmeter for the lighter gas composition.

Remember, the compressor and the associated instrumentation had originally been specified for the heavier 50 percent methane-50 percent ethane mixture. When the correct meter factor for the lighter 82 percent hydrogen gas was used, the indicated flow corresponded to the design 400,000 SCF/hr.

This story relates the performance of a reciprocating compressor to a variable gas composition. The effects are very small. However, had a fixed-speed centrifugal compressor been used in a service with such a large variation in gas molecular weight, the story would not have had a favorable outcome.

CHAPTER 43

Compressor Efficiency

Effect on Driver Load

Let's assume that we are driving a centrifugal compressor with a constant-speed electric motor. We are compressing natural gas, coming right off a thousand wellheads, in Laredo, Texas. This is not a good idea. There is entrained brine (salty water) in the gas. The brine will dry out inside the compressor case, due to the heat of compression. The resulting salts will deposit on and inside the wheels or stages of the compressor's rotor.

The compressor efficiency will be adversely affected. As a consequence:

- The flow of gas compressed will be reduced.
- The discharge temperature of the compressor will increase.
- The amp load on the electric motor driver will go down.

Why, though, does it take less work to drive the compressor when its rotor wheels are encrusted with salt? It is true that it takes somewhat more work to compress a mole of gas with a fouled rotor. But the fouled rotor also compresses a lot fewer moles of gas. Therefore, the net effect of rotor fouling is a reduced workload for the motor driver.

Let's now assume that I am driving the same compressor with a gas-fired turbine. The fuel-gas regulator to the turbine is 100 percent open. The turbine is spinning at 10,000 rpm. As the compressor's rotor fouls with salt, what happens to the speed of the turbine?

Answer—it runs faster! It is easier to spin the compressor rotor when its efficiency is impaired. The salt-encrusted wheels do not bite as hard into the gas as would clean wheels. The amount of gas moved is reduced, even though the rotor is spinning faster.

43.1 Jet Engine

A gas- or diesel-fired turbine driver is essentially the same as a *jet engine*. The burning gas spins a turbine. The turbine spins two compressors:

- The natural-gas compressor we have been discussing.
- An air compressor. The discharge pressure from this air compressor might be 80 to 90 psig. The compressed air is used as the combustion air supply to combust the turbine's fuel. The majority of the horsepower output from the turbine (perhaps 60 percent) is used to drive this combustion air compressor.

The horsepower output from a gas turbine is seldom limited by the position of the fuel-gas regulator, as I just described in the previous example. The limit is usually the exhaust temperature of the combustion or flue gases. The turbine's blades have a metallurgical temperature limit of 1100 to 1200°F (as designated by the manufacturer). The temperature of the exhaust combustion gases correlates with the temperature of the turbine blades.

Now, let us again assume that our natural-gas compressor rotor begins to foul with salt, drilling mud, and/or a paraffin wax, produced with the gas. Here is what will happen:

1. The flow of compressed natural gas will decrease.
2. The turbine and the compressor will both spin faster.
3. The combustion air flow from the front-end air compressor will increase.
4. The air-to-fuel ratio in the turbine's combustion chamber will increase.
5. The exhaust combustion flue-gas temperature will drop as the air-to-fuel ratio rises.
6. The fuel-gas regulator can now be opened, because we are no longer constrained by the exhaust-gas temperature.
7. The extra fuel gas, plus the extra combustion air, increases the horsepower output from the turbine.

Strange to say, but we could move almost as much natural gas with a dirty compressor rotor as we could with a clean compressor rotor. Of course, the amount of fuel we needed to run the turbine increased substantially as the rotor salted up. But our fuel in Laredo was self-produced, and therefore more or less free natural gas, so we did not care.

43.2 Controlling Vibration and Temperature Rise

43.2.1 Vibration

But we learned to care. You see, rotor fouling is a double-edged sword. It cuts two ways. One aspect of rotor fouling is loss of adiabatic

compressor efficiency, which wastes work and reduces flow. The other edge of the sword is vibration.

Eventually, the fouling deposits on the rotor will become so thick that they start to break off, especially if you shut the compressor down for a few hours for minor repairs to the lube-oil system. When the compressor is put back on line, bits and pieces of grayish salt break off and unbalance the rotor. At 8000 rpm, the high-vibration trip cuts off the fuel to the gas turbine, and the machine is taken off line for repair.

The compressor is disassembled. I get the opportunity to accompany the rotor to Dallas, in the back of a van that needed a new suspension. Once there, I watched the manufacturer's machinist crew clean and rebalance the wheels. I noticed that the salt deposits were thickest on the middle wheel. The last wheel was only slightly encrusted while the first wheel was clean, except for some waxy grease.

Why this sort of salt distribution? I reasoned that the entrained brine did not dry out until it reached the middle wheel. But by the time it reached the last wheel, all the salt deposits that were going to accumulate in the compressor had done so.

This gave me an idea. Suppose we injected a liquid spray into the front end of the compressor (we eventually used a heavy aromatic naphtha obtained from a local refinery). This could prevent the deposits from sticking to the spinning wheels. We tried it, and it worked. Rotor fouling and the consequent vibrations and loss of capacity became far less frequent.

This reminds me of something. Dear reader, you cannot learn anything from our book. You cannot learn about process equipment by reading about it. You have to ride in the back of the van.

43.2.2 Temperature Rise

In general, an inefficient compressor will have a high discharge temperature. As compressor efficiency declines, less of the driver's work will go into compression, and more of the driver's work will be degraded into heat.

On the other hand, a high compressor discharge temperature may be due to a larger compression ratio. The compression ratio is the discharge pressure P_2 divided by the suction pressure P_1 . Both P_1 and P_2 are expressed in psia (not in psig).

I have put these ideas together in a single equation:

$$\text{Relative efficiency} = \frac{(P_2 / P_1) - 1}{T_2 - T_1} \quad (43.1)$$

where T_2 = the compressor discharge temperature
 T_1 = the compressor suction pressure

43.3 Relative Efficiency

You cannot use the information presented in this text to design compressors. You cannot use the information presented in this book to calculate actual compressor efficiency. Those are complex subjects.

But we are not concerned with establishing the actual compressor efficiency. What we wish to know are the answers to the following sorts of questions:

- What is my compressor efficiency today compared to its efficiency last month or right after the unit turnaround last year?
- What is my compressor efficiency today compared to its design efficiency?
- What is the efficiency of cylinder A compared to cylinder B on my reciprocating compressor?
- Which is more efficient—my beat-up, old centrifugal compressor, or my brand-new reciprocating compressor? Both machines are working in parallel, but which has a better adiabatic compression efficiency?

We can answer these questions using Eq. (43.1), which defines *relative efficiency*. The calculated numerical value of relative efficiency means nothing! The equation may be used only to compare two sets of operating data. The equation is not even thermodynamically correct. But it is sufficiently correct, provided the services represented by the two sets of data are reasonably similar and the compression ratios are within 10 to 20 percent of each other.

43.3.1 Axial Air Compressor Example

An axial air compressor is like a simple, but better, version of a centrifugal compressor. Instead of wheels and stators, it has rotating blades and fixed vanes. In a modern, large jet engine, the combustion air is supplied by an axial air compressor. The machine shown in Fig. 43.1 is supplying air to a fluid catalytic cracker unit catalyst regenerator in southern Louisiana. The operating parameters shown on the sketch itself were taken when the compressor was thought to be in an abnormally fouled condition. We are going to compare the relative adiabatic compression efficiency for three conditions:

- Fouled
- Normal
- Design

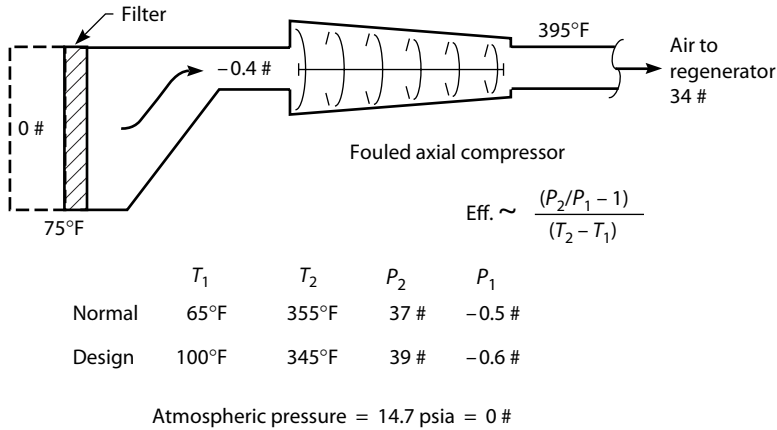


FIGURE 43.1 Example of relative compression efficiency.

To do this, we will use the relative-efficiency equation, Eq. (35.1):

- *Fouled condition:* (Data shown on sketch)

$$P_2 = 34 + 14.7 = 48.7 \text{ psia}$$

$$P_1 = -0.4 + 14.7 = 14.3 \text{ psia}$$

$$\frac{P_2}{P_1} - 1 = 2.406$$

$$T_2 - T_1 = 395 - 75 = 320$$

$$\text{Relative efficiency} = \frac{2.406}{320} = .00752$$

- *Normal condition:*

$$\text{Relative efficiency} = .00911$$

- *Design condition:*

$$\text{Relative efficiency} = .01147$$

From these relative-efficiency values, we can draw the following conclusions:

- The axial compressor running in the fouled condition is operating at only 82.5 percent of its "normal condition" efficiency. This means that the fouling problem is really rather severe.
- The adiabatic compression efficiency of the air compressor, running in its normal condition, is only 79.4 percent of its

design condition. The manufacturer's design adiabatic compression efficiency is quoted as 81 percent. Therefore, the "normal condition" adiabatic compression efficiency is approximately

$$81\% \times 79.4\% = 64.3\%$$

Not too good! A detergent solution was injected into the air intake of the axial compressor, and the adiabatic compression efficiency recovered to about 80 percent.

I like to calculate the relative compression efficiency because I do not have to know the flow of process gas. I do not have to know the driver horsepower output, the steam to the turbine, the fuel gas to the gas turbine, or the speed of the compressor. I do not have to know Z (the gas compressibility factor) or K (the ratio of the specific heats). The things I do have to know—the suction and discharge temperature and pressure—I can check with my own hands and my own tools.

43.3.2 Parallel Compressors

Let us assume that a reciprocating compressor has two cylinders working in parallel. Each cylinder has both a crank-end section and a head-end suction, where gas is compressed. In effect, we have four small compressors working in parallel. The inlet and outlet pressures, and hence the compression ratio, for each of these four minicompressors are the same. The relative efficiency for each minicompressor is then

$$\text{Relative efficiency} = \frac{1}{T_2 - T_1}$$

You can measure these temperatures with an infrared, noncontact temperature gun. True, this only tells us pipe external temperatures, but as long as we are consistent, use of such skin temperatures is acceptable.

Using this easy technique, we can guide the maintenance effort as to which reciprocating compressor valves must be overhauled and which valves are working correctly.

Caution: If the inlet side temperature for any cylinder end is hotter than the main gas inlet header pipe, then there is zero gas flow through this cylinder end. The adiabatic and relative compression efficiencies are also zero.

43.4 Relative Work: External Pressure Losses

Liz and I had a project to expand the wet-gas compressor capacity of a centrifugal machine in Pasadena, Texas. We ran a pressure survey on the compressor system, as summarized in Fig. 43.2.

Our client had been monitoring the inlet pressure to the compressor at the P_1 pressure point. But we found that the actual compressor inlet pressure was 2 psi lower, due to the in-line basket filter.

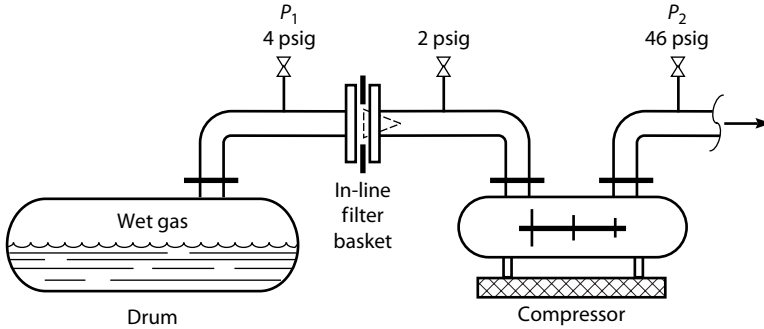


FIGURE 43.2 In-line basket filter reduces apparent compressor efficiency.

This filter had apparently never been cleaned. Our client then asked us to calculate the percent of the motor amps being wasted across the partially plugged basket filter.

To answer this question, we may use the following formula:

$$\text{Relative work} = \left(\frac{P_2}{P_1} \right)^{(K-1)/K} - 1$$

The ratio of the specific heats K for the gas was 1.33. Therefore

$$\frac{K-1}{K} = \frac{1.33-1.00}{1.33} = 0.25$$

Atmospheric pressure in Pasadena on the day our data were taken equaled 15 psia, or 30.6 Hg.

Therefore, the relative work required by the compressor, with the existing restriction of the fouled filter, was

$$\text{Relative work, fouled filter} = \left(\frac{46+15}{2+15} \right)^{0.25} - 1 = 0.376$$

The relative work required by the compressor, without the restriction of the fouled filter, would have been

$$\text{Relative work, clean filter} = \left(\frac{46+15}{4+15} \right)^{0.25} - 1 = 0.338$$

The percent of adiabatic compression work that was being wasted across the partially plugged in-line filter basket was then

$$\frac{0.376-0.338}{0.376} = \frac{0.038}{0.376} = 10.1\%$$

This seems like a rather large loss for a small, 2-psi pressure drop. But after the filter basket was pulled and cleaned and the compressor

was returned to service, the accuracy of the calculation was proved. About 10 percent more moles or volume of gas could be handled, with approximately the same amperage load, on the electric motor driver. A nice example of the Second Law of Thermodynamics in action.

If you want to be lazy and use Eq. (43.1) and ignore the fractional exponent, the work wasted would have been 11.3 percent. That is, you could just take the ratio of 19 psia divided by 17 psia. Liz calls this “Norming a problem.” Meaning it’s a quick, approximate, and lazy way of getting to the answer.

CHAPTER 44

Safety Concerns

Relief Valves, Corrosion, and Safety Trips

Your process unit likely contains a wide variety of safety features and equipment. These safety features fall under the following three categories:

- Relief valves
- Corrosion monitoring
- Alarms and trips

It is certainly my experience that the most common and catastrophic accidents in process units are related to corrosion-type failures. I cannot bring to mind any process vessels that were definitely overpressured and failed because a relief valve did not open.

Relief valves were invented to prevent steam boilers from blowing up as a result of excessive steam drum pressure. This was a distressingly common occurrence in the nineteenth century. The relief valve is also called a *safety* (or “pop”) *valve*. When the pressure in a vessel exceeds a preset amount, the relief valve is supposed to pop or spring open. Gas will then be vented from the vessel until the pressure in the vessel drops by 10 to 20 psi below its relief-valve-set pressure. We usually operate pressure vessels at 25 psig or 10 percent below the relief-valve setting.

The relief-valve-set, or pop, pressure is adjusted in a machine shop. A large threaded nut on top of the relief valve is used to make this adjustment. Air pressure is applied to the valve, and the technician adjusts this nut until the relief valve opens at the proper pressure.

The pressure at which relief valves are set to open is the vessel design pressure. The vessel design pressure (maximum allowable working pressure, MAWP) is stamped on the manufacturer’s nameplate. It is illegal to set the relief valve at a higher pressure. The vessel is probably hydrostatically tested at a pressure 50 percent greater than its

design pressure. This test pressure also is listed on the nameplate. You cannot use the test pressure as a guide to set the relief valves.

Sometimes vessels are rerated. Often, we can increase the design pressure of a vessel by taking credit for an excessive *corrosion allowance*, which is an extra thickness of metal added to the vessel wall. Designers do not include this extra thickness in their pressure-rating calculations. However, if experience teaches that there is little corrosion, then the vessel may be rerated and receive a new pressure-rating stamp. There are rigorous, formal, legal procedures to follow in rerating a vessel.

On the other hand, vessels must often be derated for age and excessive corrosion. Either way, the relief valve needs to be reset. There is only one way to know the actual relief-valve setting, and that is to climb up to the relief valve and read the tag that was fixed to the valve at the time of its last setting.

I used to reset relief valves on small pressure vessels in the natural-gas fields in southern Texas. After adjusting the nut on the valve with a wrench, I would raise the vessel pressure to check whether the valve would relieve at the proper pressure. If not, I would continue to adjust the nut and, by trial and error, find the proper relief-valve setting. This is not a particularly practical method to adjust relief valves on stream on a process unit. It is also illegal to do so.

Many relief valves pop open above or below their set point. Such valves have to be removed from the vessel and reset in a machine shop. Often, there are isolating block valves, located beneath the relief valve, that permit the relief valve to be pulled while the vessel is still in service. These block valves are perfectly legal, provided they are *chain-locked open*. It is unlawful to have an isolating valve below a relief valve that is not chain-locked open or sealed open in some positive manner.

44.1 Relief-Valve Plugging

Often, relief valves do not open because their piping connection to the vessel is plugged with corrosion products, salts, or coke. Even if this connection is only partially plugged, the effective capacity of the relief valve is greatly diminished.

A *rupture disk* is a thin sheet of metal installed below the valve, intended to protect the relief valve from plugging. The rupture disk ruptures at the relief-valve-set pressure. A better approach to retard this plugging problem is to maintain a steam purge, or inert-gas bleed, below the relief valve to prevent the accumulation of solids below the valve.

Often, relief valves fail to close once they have popped open. I hate this! I hate the tension. Will it or won't it reseal by itself? Or will some operator have to climb up the 180-ft crude distillation column and hammer on the side of the relief valve until it reseals? Especially if the relief valve is venting to the atmosphere rather than the flare, this can be a nasty and dangerous job.

Furnace tubes, process piping, and heat exchangers may also have to be protected by relief valves. Incidentally, the small 1-inch relief valves you see on many tank field loading lines and on heat exchangers are not process relief valves. They are there for thermal-expansion protection only. This means that if you block in a liquid-filled exchanger and the liquid is heated, the liquid must expand or the exchanger will fail. That is what the thermal relief valve is there to prevent.

Typically, heat exchangers, piping, and furnace tubes have a certain *flange rating*. For example, 150-psi flanges have a pressure rating of about 220 psig, and not 150 psig. A 300-psi flange has a rating of roughly 430 psig. If the upstream process pump has a shut-in or dead-head pressure exceeding this flange rating, then a downstream protective relief valve is required.

44.2 Relieving to Atmosphere

In 2004, at the B.P. Refinery in Texas City, 15 contract workers were killed and 170 seriously burned. The relief valve opened on the top of a naphtha splitter. The relief valve did not vent to the flare, but to a blowdown stack. The blowdown stack was open to the atmosphere and drained to a sewer. This drain had a loop seal to prevent vapors from blowing into the sewer.

The operators were starting the tower:

1. The tower was on total reflux.
2. The bottom level control valve was closed in the field.
3. The bottom flowmeter was off-zero, and showed a substantial flow.
4. The bottom liquid level indicator on the panel indicated a less than full level, even when the bottoms level was above the top level tap.

The operators, being unaware of the last three facts, filled the tower almost to the top with naphtha and increased heat flow to the tower. The relief valve, set for 50 psig, opened as the liquid in the tower swelled up due to the extra heat. The boiling naphtha partially flashed as it flowed from 50 psig in the tower to zero psig in the blowdown stack.

As the boiling naphtha flowed through the loop seal of the blowdown stack drain, the loop seal became vapor locked. Or perhaps the volume of boiling naphtha exceeded the drain's capacity. Regardless, the naphtha backed up in the blowdown stack to the inlet nozzle. The evolved vapors blew the liquid out of the top of the stack. The naphtha ignited in the midst of the contractor trailer park built around the blowdown stack. Two additional points are relevant:

- In the 1970s I assisted, in a minor capacity, Gary Elmer in the design of this tower. At that time, the refinery was an Amoco

Oil plant, and Gary and I worked in the Chicago Engineering Design Division. As process design engineers, we did not worry about relief valve vents.

- In 1974, I was promoted to the position of operating supervisor at the Amoco Refinery in Texas City. I had four tall distillation towers on my alkylation unit. The towers had relief valves at the top of the tower. These relief valves vented to the atmosphere. Over-pressuring was common and I saw the white plume blowing from the valves on many occasions. Never did I stop and think, "What will happen if a relief valve opens when a tower is in fully developed flood?"

Now I and everyone else in Texas City knows what happens. How about your unit? Do you have relief valves on vessels in hydrocarbon service that open to the atmosphere? What would happen on your unit if the safety relief valve opened when the vessel is full of liquid?

I have asked this question at many plants in the past years. The majority still have safety reliefs venting to the atmosphere even though everyone in the United States refinery industry knows what happened in Texas City in 2004. To summarize, when considering the safe operation of relief valve systems, the following assumptions should be made:

- The valve will open some day.
- When the valve opens, the process vessel will be full of liquid.
- The liquid will be at its boiling point at the pressure at which the relief valve opens.
- There will be a source of ignition present downstream of the relief valve.

As you evaluate the above criteria, recall that there were scores of towers in Texas City operating for many decades. These towers relieved to the atmosphere (some of which I designed), and nothing ever happened. Nothing ever happened until that terrible incident in 2004.

44.3 Corrosion Monitoring

On older process units, you may still encounter piping with sentry holes. Let's say I have a 0.5-inch thick pipe. The corrosion allowance for the pipe is 0.25 inch. A number of small holes are drilled into the pipe to a depth of 0.25 inch. When we start leaking at these small holes, this means that the pipe has corroded to its discard thickness in the area of the sentry hole. Incidentally, you can stop the resulting leak, at least in carbon steel water lines, with a brass wood screw and a screwdriver. I have also done this on hydrocarbon lines under an 80-psig pressure, but perhaps that was not too smart.

A more modern method to check for loss of thickness in process piping is by *ultrasonic testing* (UT) or Sonaray. These are portable instruments used to check pipe thickness on-stream. Do not forget, though, that the thin elbow, the one that is sure to fail, is always out of reach unless a ladder can be found. And the inspector cannot find the ladder. The outside radius of elbows are typically the thinnest portions of a piping run. I became the world's leading expert on not finding ladders and not UT-checking thin elbows at the Amoco Oil Refinery in Texas City in 1976—just prior to the alkylolation unit explosion. The explosion that wrecked half of the refinery.

44.3.1 Corrosion Coupons

I always like corrosion coupons. They are a tangible piece of hardware that you can hold in your hand. A corrosion coupon is just a piece of metal, perhaps 0.25 inch thick, 3 inch long, and 0.5 inch wide, that is inserted into the flowing process stream through a packing gland in a valve. It can be easily pulled for inspection every week or every month. The corrosion engineer weighs the coupon for metal loss and inspects it for pitting and other forms of corrosive attack. Often, a number of corrosion coupons of various metallurgy are used. The corrosion engineer then calculates, on the basis of the metal lost from the coupon in one month, the *mils per year* corrosion rate.

For example, a steel pipe is 0.5 inch thick, or 250 mils. Its discard thickness is 0.125 inch thick, or 125 mils. If the corrosion rate is measured at 25 mils per year, then the expected life of the pipe is five years. Corrosion rates in excess of 10 mils per year are normally considered excessive and unacceptable, at least in petroleum refineries.

44.3.2 Corrosion Probes

This is an electronic method used to measure corrosion rate in mils per year. The corrosion probe can be inserted through a packing gland. It is read periodically with a portable instrument that measures the change in electrical conductivity of the probe. It is simple, but perhaps a little less reliable than the coupon.

There is another sort of probe that measures ionic hydrogen penetration into and through steel walls. Such *hydrogen activity* is a product of corrosion and/or high hydrogen partial pressures in a vessel. This sort of hydrogen activity promotes hydrogen blistering and stress corrosion cracking of vessel walls. A sketch of a hydrogen probe is shown in Fig. 44.1. It consists simply of a small chamber welded onto the exterior of a vessel. A pressure gauge is connected to this chamber.

The hydrogen ions (or protons) dissolve and pass through the iron lattice structure of the vessel wall. When some of them emerge at the outside of the vessel wall, they are trapped inside the hydrogen probe chamber. There, the ionic hydrogen is converted into molecular hydrogen.

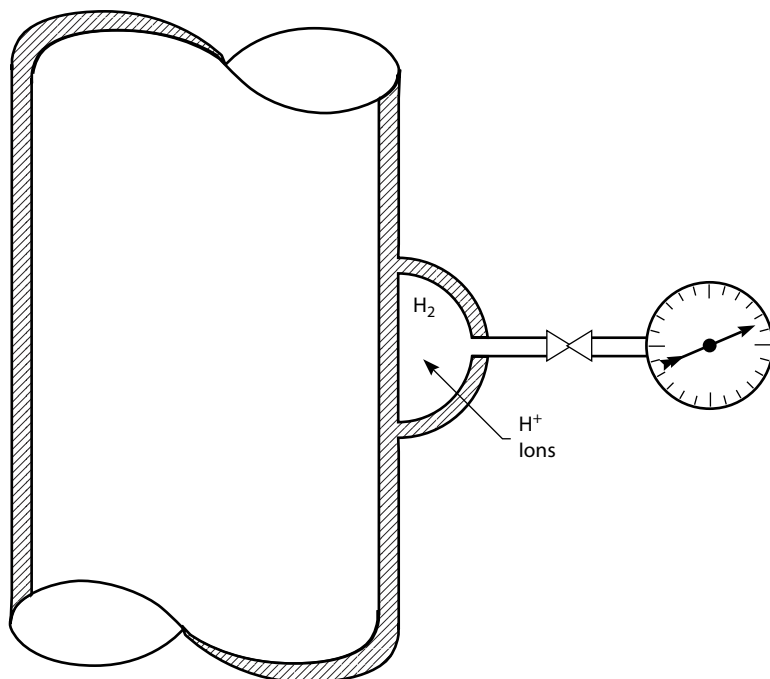


FIGURE 44.1 A hydrogen probe.

The rate of pressure increase inside the chamber is a direct measure of the amount of ionic hydrogen infiltration through the lattice structure of the steel wall of the process vessel.

Often massive corrosion failures occur suddenly, without the warning of any small leaks. Lines part at welds, vessels burst apart as a result of hydrogen-assisted stress corrosion cracking, and thin elbows peel back like the top of a soup can. Process plants are dangerous places, mainly because of corrosion; hence the importance of monitoring corrosion. It is the responsibility of the unit chemical or process engineers to monitor corrosion on their units.

44.4 Alarms and Trips

44.4.1 Safety Trips

One of the most common safety trips is the automatic fuel-gas shutoff. We have this at home on our furnaces. We ignite the pilot light manually. The pilot light heats a thermocouple. The milliamp output from this thermocouple opens the fuel-gas valve to the main burner. The gas in the main burner is ignited from the pilot-light flame. Should the

heat from the pilot light diminish below a certain point, the fuel gas to the main burner, as well as to the pilot light, will be shut off.

A less common type of fuel-gas trip to a heater is a low-pressure trip. A pressure transducer generates a milliamp output from a boiler feedwater pump. Should this milliamp output fall below a certain level, the instrument air signal to the fuel-gas regulator actuator will be shut off. These fuel-gas valves are *air-to-open*, meaning that loss of instrument air flow causes the valve to close.

Some fired heaters, especially boilers, have a device called a “purple peeper,” which is simply an optical device that looks at a flame. If it does not detect light with a wavelength in the high-frequency (i.e., purple) end of the optical scale, it interprets this as a flame-out. The fuel-gas regulator is automatically shut.

High process heater outlet temperatures are another trip point for many heaters. These sorts of trips are subject to a rather deadly malfunction. If the process flow is totally lost, the heater tubes may become extraordinarily hot. But the high-temperature trip may be located too far from the heater outlet to be affected by the high temperature in the heater. This particular malfunction occurred in a southern Louisiana refinery, and indirectly led to the deaths of a score of workers. This is why we have P&ID (piping and instrumentation diagram) reviews when our unit is being designed. Fuel to a boiler may also be shut down by a low water level in the boiler’s steam drum. The low-level trip is simply a float chamber. If the float drops to the bottom of the chamber, it flips a *mercuroid switch*, which shuts off the fuel-gas supply to the boiler. These mercuroid switches look quite similar to the thermostat switch we have inside our house, which switches the air conditioner on or off. Note that these level trip float chambers, even though they are quite short, still require two level taps on the vessel.

Some heaters also have a low fuel-gas pressure trip on the fuel gas itself. The idea here is that if fuel-gas flow is lost, we do not want it to surge back into the heater too quickly if the fuel-gas pressure is suddenly restored.

44.4.2 Compressor Trips

Our home circuit breakers, or fuses, are, of course, trips to prevent overheating electric circuits or electric motors. The only difference is that at work our electric circuit breakers have a built-in time delay. This is needed to allow the motor driver to overcome the starting torque inherent in most large pieces of rotating process equipment.

Also, the motor trips are activated, not by excessive motor temperatures, but by an excessive measured amp load to the motor. This is called the *FLA* (full limit amp load) point. The FLA point may be increased by perhaps 10%, but this will reduce the motor’s operational life. Our motor trips at home are activated not by the measured amp load, but by excessive motor temperatures.

Compressors also have vibration trips. These trips measure the amplitude of the vibrations—which, if they become excessive, will shut off the fuel, steam, or electricity to the compressor’s driver.

Turbines, both gas and steam, also have overspeed trips. These consist of a little flywheel constructed from three balls. The little balls are spread apart by centrifugal force. The greater the rpm, the greater the centrifugal force. If the balls spread apart too far, they activate the trip. We have James Watt to thank for this neat innovation, still used in its original form.

Most large compressors also have a low-lube-oil-pressure trip. This again would shut off the fuel or steam to a turbine if the lube-oil pressure gets too low. It is interesting to note that the low-lube-oil pressure that activates the trip is not the lube-oil pressure to the bearing that would be damaged because of a lack of lube-oil flow. Rather, it is the low-lube-oil pressure to the trip switch itself that directly shuts down the turbine or motor. Thus, with a low-lube-oil pressure, one can trip off a compressor without actually losing lubrication flow to the bearings at all.

Compressors are also served by high-liquid-level trips in their upstream knockout drums. These high-liquid-level trips work in the same way as the low-level boiler trips discussed above, except that the mercurioid switch is activated by a rising, rather than a falling, liquid level so as to protect the compressor from a slug of liquid.

A final word about trips. Any trip that is not tested on some routine basis can never work in an emergency. I will guarantee you that when the coupling shears on that steam turbine driving that 4000-bhp compressor, the overspeed trip that should shut off the steam flow to the turbine will not in fact trip if you have not tested that same trip recently. I promise you that the trip valve mechanism will be encrusted with hardness deposits from the steam. While the trip lever may be unlatched, the turbine will continue to spin merrily along, until it self-destructs as a result of uncontrolled overspeed. And, ladies and gentlemen, you may imagine how I have become so knowledgeable about this particular subject.

44.5 Auto-ignition of Hydrocarbons

Heavy hydrocarbons, such as vacuum tower bottom products, tar, and pitch, are frequently pumped and stored in tanks above their auto-ignition temperatures. The process reason for this is to prevent these fluids from “setting up” or solidifying in the pipelines or tanks. At ambient temperatures, even in hot climates, these fluids would normally be either solid or extremely viscous.

A hydrocarbon above its auto-ignition temperature will spontaneously ignite upon exposure to air without any source of ignition. This is why there are often fires on pumps that pump heavy hydrocarbons. If the seal on a pump in such a service begins to leak, the

heavy material then leaks out and spontaneously catches fire. Be careful when taking pressure surveys or sampling from these hot, heavy hydrocarbon lines to avoid spillage. The risk of auto-ignition increases for fluids above their auto-ignition temperature as the surface area of spillage increases. As the surface area increases, the surface area of the hot, heavy hydrocarbon exposed to the oxygen in the surrounding air also increases. Spillage on the ground, leaks into insulation material, and spillage onto rags are all increasing the risk of auto-ignition by increasing the surface area exposed to oxygen.

44.5.1 A Hazardous Piping Configuration

Sections of piping that contain a pair of alternate 90-degree elbows or alternate right-angled bends are hazardous. These should be avoided when designing pipe work because:

- When fluid flows through such pipe work, it changes momentum at the opposing angles of the turns that the pipe work requires it to make. These changes in momentum act upon the pipe work configuration to make it rotate or spin, just like a two-sailed windmill where the right-angled ends are the “sails,” or just like one of those fireworks that come as a pair of identical fireworks mounted in opposite directions at either end of a piece of wood—we pin the wooden strip to a fence, light the pair of fireworks, and watch the assembly spin. Piping sections are anchored, but considerable stresses are set up within such a system.
- The changes of momentum occurring in the fluid as it flows around the bends make the inner wall at the outer edge of the elbows particularly susceptible to erosion.

Outages and some serious incidents have occurred as a result of the use of this type of piping configuration (see Fig. 44.2).

44.6 Paper Gaskets

I was talking to my friend Dan Rogers at a Texaco refinery. Dan said to remind my readers to beware of “paper gaskets.” Gaskets seal the ends of flanges used to assemble piping spool pieces. High-temperature and high-pressure gaskets are made of metal. In the past few years, the use of so-called “paper gaskets” has become common. Theoretically, these gaskets have a temperature rating of 500°F. In practice, they should only be used for:

- Blinding equipment during a unit turnaround when the blind flange will not be exposed to much more than ambient temperatures

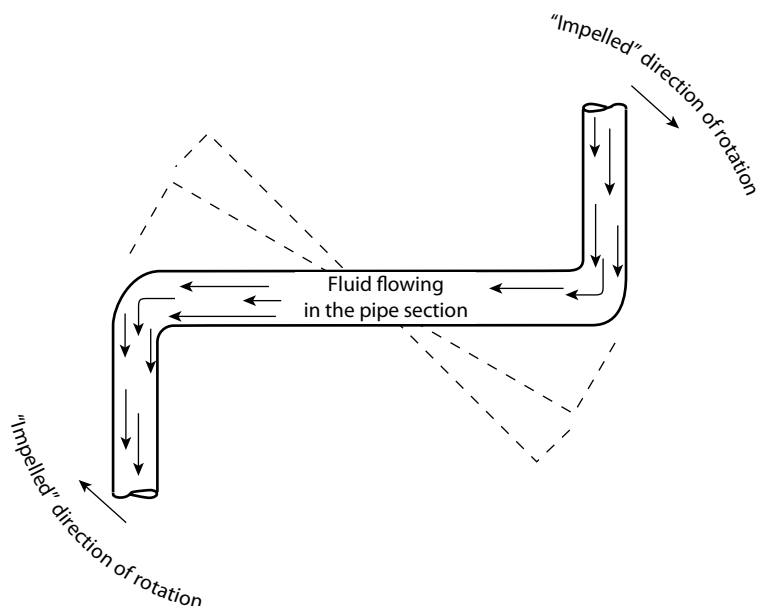


FIGURE 44.2 A hazardous piping configuration.

- Cooling water, plant air, firewater, low-pressure steam
- Services below 250°F or where a leak will be an inconvenience and not a safety issue

These paper gaskets will look pale green-blue when viewed edge-on between the flanges. Dan says that their use has caused several fires at his plant for services below their rated 500°F temperature limit.

44.7 Calculating Heats of Reaction

My client had burned a hole in a hydrotreating reactor at the Coastal Refinery in Aruba. The reactor was not in service, it was on standby with once-through hydrogen flowing through the reactor. The hydrogen was coming from a hydrogen production plant, which was in a start-up mode of operation. The failure in the side of the reactor looked like someone had burned a hole through the 4-inch steel wall from inside the vessel. This was consistent with plant data that indicated reactor temperature had exceeded 1800°F.

The failure was due to combustion of oxygen plus hydrocarbons. The combustion reaction involved was:



This is a methanation reaction. The hydrogen reacts with the carbon monoxide to produce water, methane, and heat. My job was to calculate how hot the reactor could get with 4 percent carbon monoxide in the hydrogen supply gas. To do this I looked up in Perry's *Chemical Engineering Handbook*, McGraw-Hill 5th ed., 1973, the heats of formation of:

- $\text{CO} = 24.21$ kcal per gram-mole
- $\text{H}_2 = 0$ (as are all elements)
- $\text{CH}_4 = 17.88$ kcal per gram-mole
- $\text{H}_2\text{O} = 15.80$ kcal per gram-mole

Adding up the heats of formation on both sides of Eq. (44.1) and taking their difference gives a total heat of reaction of 49.2 kcal (49,200 calories). This would be the heat of reaction if there were 25 percent CO and 75 percent H_2 in the feed gas. However, the feed gas contained 4 percent CO, not 25 percent CO. Therefore, the heat of reaction was:

- (4 percent divided by 25 percent) \times 49,200 = 7900 calories

The 96 percent hydrogen and 4 percent CO in the feed gas would be converted to:

- 92 mole percent hydrogen
- 0 carbon monoxide
- 4 mole percent methane
- 4 mole percent water

The combined specific heat of this mixture is:

- 7.0 calories per mole per 1°C

Dividing the heat of reaction (7900 calories) by the specific heat of the products of reaction (7.0 calories per mole per 1°C) results in a temperature rise of 1130°C or 2040°F .

This calculation shows how 100°F hydrogen flowing to the reactor could have reacted, when exposed to a catalyst bed, to produce temperatures of 2000°F .

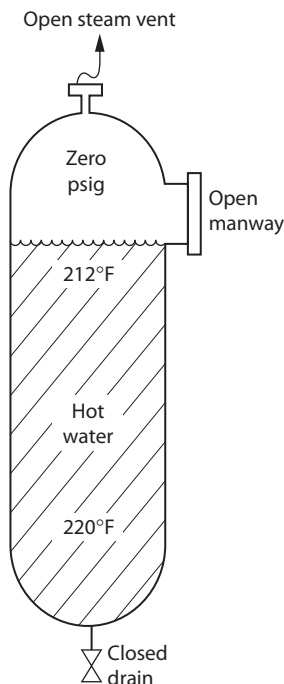
44.8 Hot Water Explodes Out of Manway

I just put down the phone. Two pipe fitters were burned opening the demineralizer drum shown in Fig. 44.3. My client wanted to understand what happened. It's a common incident.

The water-filled drum had been depressured by venting to the atmosphere. The drum had been drained to a level just below the manway. The manway was carefully and slowly unbolted and removed.

FIGURE 44.3

Boiling water can explode out of a manway.



Nothing happened for a few minutes. Suddenly the hot water boiled out of the open manway.

This happens because the drum is not at a uniform temperature. The water in the drum is at its bubble point or saturated liquid temperature throughout the drum. But because the water in the lower portion of the drum is at a higher pressure due to the static head of water, its saturation temperature is also higher than the water at the top of the drum. Also, the cooler water at the top of the drum is a little denser than the hotter water at the bottom.

If something happens to disturb this delicate balance, the hot water at the bottom will move toward the top. As the hotter water moves up, it loses a bit of head pressure and starts to vaporize. The bubbles of steam stir up the vessel's contents and accelerate boiling and steam evolution. The sudden generation of steam pushes the hot water out the open manway.

Honestly, this explanation is my best guess. But what I'm not guessing at is that this incident is not unusual. It's safer to drain saturated water levels well below manways before they are opened. I have accidentally drained 180°F heavy naphtha on my gloved hand without injury. A similar careless incident with 180°F water resulted in a trip to the Texas City infirmary. Excluding fire, hot water is more hazardous to personnel than hydrocarbon liquids of the same temperature. Treat hot water with care and respect.

CHAPTER 45

Relief Valve System Design

The function of a relief valve is to protect a vessel, piping system, or heat exchanger from exceeding its maximum allowable working pressure (MAWP). The MAWP is shown on the code stamp (a metal plate) attached to each process vessel, and by design is determined by the size of the impeller on the pump upstream of the vessel (see Sec. 34.4.1 in Chap. 34). Normally, the relief valve is set to open at this MAWP pressure. The relief valve setting is adjustable and is done by a machinist at a shop pressure-testing facility.

Once a relief valve is forced open by exceeding its set pressure, fluid flows out into the relief valve header piping. In most plants, there is a piping system that collects the effluents from various relief valves. This collection system, in turn, vents to the plant flare system.

Only too often, the relief valve, after opening due to excessive pressure, does not close properly when the pressure in the vessel drops below its MAWP. We say that the relief valve has “not reseated.” Typically, I will bang violently on the valve to encourage it to reseal. Often, it will continue to leak anyway and then a complete unit shut-down may result, so as to repair the faulty relief valve.

As a process design engineer, I do not size relief valves. This is easily done by a computer program based on the vessel size, fluid properties, and maximum operating pressures. The location of the relief valve is a matter, however, that requires careful engineering judgment. I will illustrate the problem by several examples drawn from my own refinery experience.

45.1 Coke Drums

It was standard practice to protect refinery delayed coker, coke drums, from overpressure by locating a relief valve on the overhead vapor line. However, the main reason the coke drums would overpressure was coke accumulation in the inlet of the vapor line, immediately downstream of the coke drum itself and typically just upstream of the relief valve. Thus, it was a common occurrence (for example, at the

Suncor Coker at Ft. McMurray and the Citgo Coker in Lake Charles) for the coke drums to overpressure without the relief valves opening. In one instance, the drum pressure reached over 100 psig, even though the relief valve was set at 60 psig.

Now, it is standard practice to site all coke drum relief valves on the coke drum itself, and never on the vapor outlet line.

45.2 High-Pressure Fixed-Bed Reactors

Most fixed-bed reactors are designed for downflow. I have in mind a typical multibed hydrotreating gas oil reactor or a hydrodesulfurizer diesel oil reactor. The catalyst bed supports are designed to support the weight of the catalyst plus the downward force of the catalyst bed pressure drop.

If the relief valve of a high pressure reactor opens, and the relief valve is located on top of the reactor, then the sudden upward surge of hydrogen-rich gas will lift up and upset the catalyst bed support and the interbed vapor-liquid distributors.

This exact scenario transpired at a 3000-psi white oil hydrotreater reactor I had designed for the Amoco Oil Refinery in Whiting, Indiana. All seven catalyst beds were destroyed, along with my patented reactor interbed quench distributors. The place to locate the pressure relief valves is not at the top, but at the bottom, below the final catalyst bed support of the reactor.

45.3 Trayed Towers and Packed Columns

The same logic may apply for tower trays. Of course, such trays are designed for vapor upflow, and if a relief valve on the top of the tower opens, ordinarily this will not upset the trays. Still, for higher pressure towers of larger diameter, sized with larger relief valves, a potential problem exists. If opening the pressure relief valve causes the vapor flow through the trays to increase by a factor of 3, then this would likely damage the tray deck integrity due to the sudden uplift vapor surge.

This logic would also apply to packed towers. However, designing for a strong bed limiter (i.e., packing a holddown grid) would largely avoid this sort of failure.

I have never seen or heard about tray failure due to a relief valve opening. However, in a packed distillation tower, I am aware of at least one such incident, also at the Whiting Refinery.

45.4 Liquid-Filled Vessels

Relief valves located on liquid-filled reflux drums can be a major safety problem depending on the design and size of the flare knockout (K.O.) drum. Some plants have giant flare K.O. drums with large capacity

automatic pumps. In some plants, the K.O. facility is integral with the flare stack itself and the pumps are manually operated.

Overfilling a flare K.O. drum is a serious matter. Liquid hydrocarbons may erupt from the flare like a giant flame-thrower. My secretary ran out of the Good Hope Refinery without her shoes when our flare stack overflowed. Burning liquid butane erupting from the Amoco Oil flare in Whiting, Indiana, was reported as a missile launch by a spy satellite.

The ASME Boiler Code may require a liquid-filled vessel to have a pressure relief valve if it can be isolated from another vessel that has relief valve protection. Then the designer's obligation is to determine if the flare system can handle the volume of liquid depressuring through the relief valve on the liquid-filled vessel.

45.5 Sour Water Strippers

45.5.1 Relief Valve Effluent Disposition

In 2006 a terrible fire and explosion killed 15 people and injured 500 other contractors and employees at the BP Refinery in Texas City. The main cause of the disaster was a relief valve on a naphtha splitter that vented to the atmosphere via a blowdown stack rather than to the flare.

The theory for the location of the relief valve location and disposition was, I imagine:

1. Locate the relief valve on the overhead vapor line.
2. If the relief valve opens, only vapors would be emitted, as only vapors are present at the top of the tower.
3. Hydrocarbon ambient emissions during a plant upset were legal.

I have discussed the entire incident in detail in my book, *Troubleshooting Process Operations*, 4th ed. (PennWell, 2009). The main error made in the design was the assumption of only vapor at the top of the tower. What was to happen if the tower was completely flooded, in this case due to their level instrumentation output?

After the accident, I discussed with BP engineers whether they had learned anything from their tragic experience. They responded that they no longer connect vessel relief valves in hydrocarbon service to an atmospheric vent.

But is this really true? For example, I discussed the design of an existing sour water stripper with BP engineers and operators. As this was a water stripper, the relief valve had not been connected to the flare.

However, in refinery sour water stripper feed tanks, I always find a layer of light hydrocarbon floating on the top of the sour water. If these lighter hydrocarbons are accidentally charged to the stripper

(which happens frequently), then the tower will overpressure, and the relief valves will open. Thus, reproducing the failure at Texas City.

Also, the emissions of uncombusted hydrogen sulfide vapors (H_2S) to the atmosphere could also be (although more remotely) a potential safety issue. This happened at the Good Hope Refinery where I worked in 1981, when reports of noxious odors were reported from a local high school. Our plant manager appeared on the television news to assure the community that while the H_2S odor was bad, it was not in the least harmful to the children.

Incidentally, having higher pressure reactors, or other vessels, relieving into lower pressure fractionators is not always a good design practice. I just put down my phone. I had been talking to the unit engineer at a U.S. midwestern delayed coker. They had overpressured a coke drum during water quenching. The relief valve was vented into the coker fractionator. This had apparently damaged some of the trays in the coker fractionator. A better choice might have been to connect the coke drum relief valves to the coker's blowdown system.

45.6 Protecting Relief Valves from Fouling and Corrosion

There are two methods that I have used to protect process relief valves from exposure to corrosive or fouling environments:

- Rupture disks
- Purge steam or gas

Rupture disks are thin sheets of metal that will fail or rupture if the relief valve itself opens. The disk acts as a barrier between the relief valve and the process fluid. The problem with rupture disks is that as an operating manager, I was never sure whether the rupture disks were actually intact for any particular relief valve. But what I could be sure of, once a relief valve had opened, was that the rupture disk protection was gone and could not be restored without shutting down and pulling the relief valve.

The use of a vapor purge under the relief valve is a more practical approach to protecting relief valves from plugging. However, if a purge gas such as nitrogen or natural gas is used, then a non-condensable vapor will be introduced into the process stream. Steam is often preferred for this reason. But then the possibility exists of getting a slug of water into the process vessel. The water may then flash, thus causing a pressure surge and possible vessel internal damage.

45.7 Dual Relief Valves

One practical method to maintain relief valve operability is to design the relief valves for on-stream cleaning and maintenance. As an operating manager, this was my preferred design for relief valves subject to fouling.

Of course, an isolation gate valve was required underneath each relief valve. This is legal as long as the gate valve is chain-locked open. The bigger problem is that if one relief is removed from service for maintenance, then the remaining relief valve, or valves, must still have full and adequate, capacity. Typically, for larger vessels there are three 50 percent capacity relief valves.

45.8 Process Design Responsibility for Relief Valve Design

As a process engineer, I consider it my responsibility to specify:

- The location on each process vessel of the required relief valves
- Where the relief valve should vent to (atmosphere, blowdown, or plant flare)
- Spare relief valve, purge vapor, or rupture disk to maintain the relief valve operability in a fouling service
- Range of the fluid pressure characteristics in the vessel being protected

Quite often, a higher pressure vessel will vent, not to a flare, but to a lower pressure vessel. For example, a high pressure reactor might relieve to a low pressure fractionator to recover the liquid portion of the relief valve effluent in the larger fractionator. But then the fractionator relief valve must be sized for the vapor venting rate from the high-pressure reactor. A complex engineering problem results.

45.9 Relief Valve and Pressure Sensing Connections

Figure 45.1 shows the design of my alkylation unit depropanizer relief valve at the Amoco Texas City Refinery in 1975. The tower pressure control was on steam flow to the reboiler.

I had been injecting a corrosion inhibitor into the depropanizer with the reflux. The inhibitor was supposed to be soluble in liquid propane, but it was not. After several months of using this inhibitor chemical, the relief valve nozzle of the top head of the depropanizer plugged up. But how did this first come to my attention?

1. When the relief valve nozzle plugged, the PRC (tower pressure control) began to show a lower operating pressure.
2. The steam flow to the reboiler increased (i.e., heat makes pressure).
3. The PRC setting was at 270 psig. However, my shift foreman, Bobby Felts, observed on a local pressure gauge at the tower bottom gauge glass, a far greater pressure, of 400 psig.

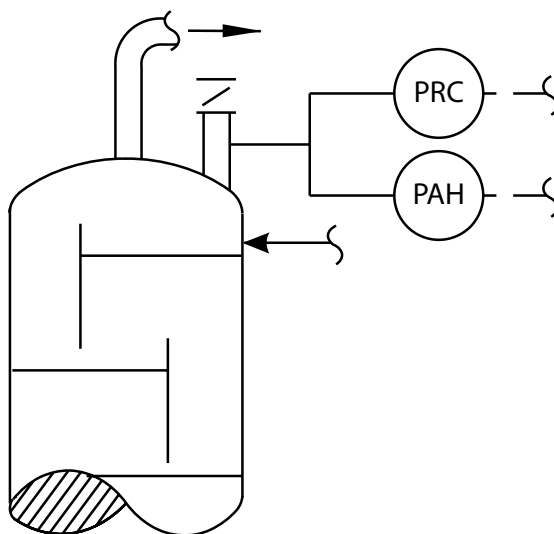


FIGURE 45.1 Do not locate PRC/PAH on relief valve nozzles.

4. The PAH (high pressure alarm) shown in Fig. 45.1 did not alarm, as it was also connected to the plugged relief valve nozzle.
5. Finally, the pressure safety relief valve failed to open, even though it was set to open at 300 psig.

Carbon steel vessels are more tolerant of this sort of abuse than chrome steel vessels, which are more likely to rupture, especially at lower pressures. I suppose otherwise, something terrible might have happened.

The advantages of locating the high pressure alarm nozzle (PAH) and the pressure controller (PRC) on their own dedicated $\frac{3}{4}$ -inch connections is that each device can be individually isolated, tested, and its nozzle cleared by the outside operator without interfering with the function of the relief valve. I have discussed these procedures in detail in my book, *Troubleshooting Process Plant Control* (Wiley, 2009).

What, you may wonder, did I do when I discovered that my depropanizer relief valve connection was plugged and that I had lost the over-pressure protection for this vessel as a result? I should have shutdown the depropanizer. But this was a 25,000 BSD alkylation unit, the world's largest alky unit. So I said nothing and continued to rely upon the principle that God protects the feeble-minded, and just kept running.

45.10 Heat Exchanger Safety Reliefs

Process heat exchangers are ASME coded pressure vessels with a MAWP (Maximum Allowable Working Pressure), just like any other vessel, for both the shell and tube sides. However, they normally are

not protected from overpressuring by a relief valve. The small relief valves you may see located on the shell and channel head (tube side) are for thermal relief, not for excessive pressure due to accidentally exposing the exchanger to excessive upstream pump discharge pressure (perhaps because a pump impeller was oversized). Such exchangers are legally protected from excessive pressure by their flange rating, and not by relief valves.

If you did not wish to provide the extra mechanical strength (i.e., the extra MAWP), then relief valves are required for heat exchangers. The same logic also applies for process piping. For instance, 150-psig piping has an MAWP of about 225 psig, depending on its temperature.

45.11 Relief Valve Effluents

The effluents from my sulfuric acid alkylation unit were strictly segregated:

- One system contained dry (water-free) acid-contaminated hydrocarbons.
- The second system was wet (acid-free) water-containing hydrocarbons.

The objective was to prevent corrosion in the relief valve headers flowing into the dual flare K.O. drums. I was always very careful, when adding new process services to the relief valve system, to adhere to this successful segregation scheme.

The designer should consider, when configuring the tie-ins to a relief header, any possible reactions between the effluent streams. On my alky unit, the reaction between strong H_2SO_4 (which is noncorrosive) and acid-free H_2O will produce terribly corrosive weak H_2SO_4 , which will destroy the welds in carbon steel piping in a matter of just a few days.

45.12 Maintaining Flare Header Positive Pressures

It is certainly required to maintain a positive pressure throughout the relief valve effluent piping system to preclude the entry of air, which, at least theoretically, could cause an explosion.

On many occasions in refineries, I have measured such pressures, which vary between perhaps 6 to 20 inches of water. There are two ways of maintaining such a positive pressure. The smart way and the dumb way.

45.12.1 Flare Water Seal

I'm sure you have seen the flame of a flare surging. That is not typically due to a relief valve opening periodically. It is due to periodically blowing the water seal in the flare seal pot (often located integral with

the bottom of the flare stack). Using a long water bottle and a clear length of plastic tubing, I have monitored the variability in the back-pressure from the water seal between 6 to about 18 inches of water. The 18 inches of water roughly correspond to the depth of the water seal, which holds back-pressure in the flare system.

At the request of one VP of refining in India, who found the flare flame surges objectionable, I reduced the depth of the water seal, and the intensity of the flame surges were thus reduced. But, of course, not the amount of hydrocarbons consumed.

Maintaining a water seal of about 10 inches of water in the flare seal drum is the smart way of preventing air intrusion into the relief valve collection system.

45.12.2 Flare Header Purge Gas

It is not uncommon for my clients to maintain a positive pressure in the flare system by continuously purging the most distant ends of the flare header piping with fuel gas, natural gas, or even propane. This is an expensive alternate to the water seal method I described above. It eliminates the need for the water seal and also the annoying sight of the pulsating flare flame. But to me it seems like a wasteful and, hence, dumb alternate.

But what I find even worse is the use of both options together. For what purpose, I cannot imagine. At least in every refinery I have worked in, there were sufficient relief valve leaks and unreported venting of hydrocarbons to keep the flare system under a higher than desired back-pressure. But, even without such leaks, the flare system would still be maintained positive just with the water seal.

In conclusion, please do not use both hydrocarbon purges and the flare stack water seal together. One or the other! Using both does not enhance plant safety and makes no engineering sense. I tried to explain all this to the refining VP referred to above, but to no avail.

45.13 Leaking Relief Valves

Many relief valves leak hydrocarbons to the flare. If the vessel with a leaking relief valve is operating at a substantial pressure (about 30 or 40 psig), then the line downstream of the relief valve will be noticeably (perhaps 10°F) colder than the relief valve connection itself, due to a Joule-Thompson expansion.

If the vessel with the leaking relief valve is liquid full with light hydrocarbons (ethane, propane, or butane), then a leaking relief valve will be quite obvious. The line downstream of the faulty pressure relief valve will be covered with ice. The ice is atmospheric moisture freezing on the line to the flare. This is not due to a Joule-Thompson expansion, but due to the auto-refrigeration of the volatile hydrocarbon liquid as it flashes to a vapor in the flare header piping.

45.14 Tray Failure Due to Relief Valves

Relief valves are often placed toward the bottom of distillation towers. In particular on crude oil fractionators, relief valves are commonly located in the flash zone (i.e., at the feed elevation).

Many such columns are equipped with movable valve caps. If the relief valves open, the vapor flow through the trays will be reversed. This will seal the valve caps up against the tray panels. The large downward force exerted by the trapped vapors could then collapse the trays.

For such a tower, tray decks should be sieve or grid trays, and never valve cap trays. I will typically use a grid tray with a ½-inch lift. Also, using 10-gauge rather than 14-gauge trays (which is the industry standard) will double the tray resistance to collapse, due to the relief valves lifting at the bottom of the tower.

45.15 The Piper Alpha Rig Destruction

The Piper Alpha Oil Platform in the North Sea, about 100 miles from Aberdeen, Scotland, exploded and was destroyed by fire in July 1988. One hundred and sixty-five people were killed. I was hired by Occidental Petroleum, the operator of the platform, to write an independent report of the catastrophe. The basic cause of the incident was clearly the release of light hydrocarbons (propane, butane, pentanes) associated with a condensate flash drum located on the production platform.

Occidental's investigation indicated that the accident was the result of two (now dead) pipe fitters not replacing a relief valve properly on a pump discharge line. My report was contradictory to Occidental's conclusion and consisted of the following observations:

- A level instrument malfunction and/or operator error had resulted in a high liquid level in the separator vessel.
- The vessel overpressured and the relief valve opened.
- Light liquid hydrocarbons saturated with water (and perhaps some free water) flashed across the relief valve.
- The resulting auto-refrigeration caused the pressure relief line to the flare to plug with hydrates. Note that hydrates (an ice-hydrocarbon solid) will form at temperatures well above the freezing point of water.
- With the flare line plugged, the vapor-liquid separator vessel overpressured and failed (i.e., possibly a flange blew out).

To support my conclusion, I noted that the exterior of the flare line had been observed covered with ice shortly before the fire and that a high liquid level had been noted in the vessel that day. I submitted my report to Occidental, received payment by wire transfer, and never heard another word about my report from my client.

This week I'm teaching at the Sasol coal-to-liquid plant in South Africa. They related an incident pertaining to a pressure relief line plugging with hydrates, due to steam leaking into a flare header, which contained liquid propane and propylene that were also leaking through a defective relief valve. It reminded me of my report to Occidental pertaining to the Piper Alpha catastrophe all those long years ago. The lesson is that relief valves in light (propane and butane) liquid hydrocarbon service, where moisture may be present, create a potential for disaster due to the relief valve line totally plugging with hydrates.

CHAPTER 46

Corrosion— Process Units

What are the two most powerful corrosive agents in a process plant? Not acids or sulfur, or caustic or salt. Neither cyanides nor carbonates nor chlorides. The most aggressive corrosive agents are air and water. Nitrogen is inert. It is oxygen in an aqueous environment that is the main cause of many corrosion problems.

Rusty carbon steel pipe is the commonly encountered form of corrosion due to air and water. Stainless steel appears to be immune to such corrosive attack, but this is not quite true. Stainless steel contains a substantial concentration of chrome and nickel. Common 304 stainless steel has 18 percent chrome and 8 percent nickel.

The chrome especially is subject to oxidation when exposed to wet air. But the metallic oxides so formed adhere tightly to the surface of the metal. This rugged metallic oxide layer acts as a barrier to prevent further destructive oxidation of the base metal. The oxide layer is so thin that it is smaller than the wavelength of visible light. Thus, it is quite invisible to the human eye. Stainless steel piping appears shiny after years of service, even though its surface has been oxidized after just a few minutes of service.

Anyone who has ever painted a wrought iron fence to prevent rust appreciates the concept of a protective barrier. Aluminum and titanium are protected from corrosion due to the rapid formation of a surface metallic oxide layer, which tightly adheres to the metal surface.

The problem with ordinary carbon steel is that the oxide formed does not stick very tightly to the metal surface. The essence of corrosion control then is to manipulate process parameters so that a protective barrier of products of corrosion does adhere to the surface of carbon steel vessels and piping.

46.1 Closer to Home

I've just spent \$8000 replacing my central air-conditioning system at my home in New Orleans. The outside condenser coils had corroded. Soil built up along the bottom row of tubes. The wet soil plus air over a 10-year period had caused a freon leak.

The evaporator coils in my attic were also leaking. The condensate collection pan had not drained freely for the past five years. Condensate had backed up and submerged the bottom tube of the evaporator and caused the tube to corrode and leak freon.

Lesson: Keep the outside of your home air-conditioner heat exchangers free of dirt and water, and the evaporator and condenser coils will last for 20-plus years.

46.2 Erosive Velocities

High-fluid velocity will tend to erode off the protective corrosion layer inside process piping. This exposes new, clean, and reactive metal to the corrosive oxygen and water. What is meant by erosive velocity? For aqueous systems with a few hundred parts per million (ppm) of particulates, an erosive velocity is

$$\text{Erosive velocity (feet per second)} = \frac{65}{[\text{Density}]^{0.5}}$$

where density is expressed in pounds per cubic foot.

For example, dirty water (which has a density of 62 lb/ft³) would become erosive at about 8 ft/s. We sometimes design water pipes for velocities of 10 to 12 ft/s. That's fine, provided the water is clean and noncorrosive. For dirty water, a design velocity of 5 to 7 ft/s would be better.

46.3 Mixed Phase Flow

When dealing with a vapor-liquid mixture, I assume the flowing velocity and density to be an average of both phases. As long as their combined velocity is above 15 ft/s, we can safely assume both phases are moving in what is called emulsified flow through the piping.

But what happens if a saturated liquid suddenly starts to boil inside a pipe? This is called *cavitation*. For example, hot water saturated with air suddenly enters a section of smaller diameter piping. The resulting increase in velocity causes a reduction in flowing pressure. The reduced pressure causes that water to boil violently. The turbulence created by the generation of bubbles of steam combined with the oxygen cause rapid corrosion due to a combination of localized oxidation and localized erosion.

Be careful when hot water that has not been deaerated is used in a process plant. Cavitation failures in hot water piping systems are a common problem.

46.4 Carbonate Corrosion

My mother still thinks I'm very smart. Forty-eight years ago, I removed rust stains from our bathtub with Coca-Cola. It is the carbonic acid in Coke that removed the iron oxide rust stains from the tub. This is an illustration of the corrosive nature of carbonic acid brought into contact with ferric compounds. The carbonic acid is formed from CO_2 dissolved in water.

I mentioned an example of how CO_2 becomes concentrated in steam systems (see Fig. 12.6). Anytime CO_2 starts to dissolve in water, carbonic acid will form, which has a relatively high pH as compared to other common minerals acids (sulfuric, hydrochloric, nitric acid). Actually, carbonic acid is aggressively corrosive to carbon steel at a pH of 5.5. Sulfuric acid at a pH of 5.5 is only mildly corrosive to carbon steel.

CO_2 is often produced by the chemical breakdown of calcium carbonate or sodium carbonate. CO_2 is generated as a by-product of the production of hydrogen and ammonia. Processing of natural gas for pipeline sales is another common source of CO_2 . The effluent from a sulfur plant or any combustion process contains wet CO_2 .

46.5 Naphthenic Acid Attack

Most crude oils contain some naphthenic acid. These acids tend to concentrate in those crude oil components that boil between kerosene and heavy gas oil. Naphthenic acid corrosion is characterized by a general thinning of stainless steel vessels and tower internals at temperatures between 450 to 750°F. Rates of corrosion are greatly accelerated in areas of high velocities. Naphthenic acid corrosion is retarded by H_2S . Thus, process equipment containing hydrocarbons with higher sulfur contents are, to some degree, protected from naphthenic acid corrosion.

My experience indicates that stainless steels that have high chrome and nickel content are not very resistant to naphthenic acid attack. What is required is molybdenum. 304 stainless steel is unsuitable due to its low moly content. Even 316 stainless steel, which has about 2 percent moly, offers only marginal protection. I usually specify 317 stainless steel, which has about 3 percent moly, when designing for naphthenic acid services.

46.6 A Short History of Corrosion

From an operator's perspective, corrosion appears to be a relentless, unavoidable, and gradual process. Pipes do rust and eventually fail, but the rust takes years to cause a failure. It is rather like cigarette smoking—clearly a deadly habit, but very slow to progress to disease.

But this is not always the case. Corrosive failures sometimes appear suddenly. I have described below a few incidents from my experience to illustrate that corrosive failures can occur in a matter of hours, or even minutes, in the right (i.e., dangerous) environment.

46.6.1 Weak Sulfuric Acid Corrosion

It is 3:00 A.M. and I am fast asleep. The ringing phone drags me out of bed. It is Henry Zipperian, my not-so-wonderful shift foreman.

"We got a butane leak, Norm, on the reactor effluent line."

"Well, what do you expect me to do about it? You're the night-shift foreman, Zip. Just handle it. Just don't bother me."

"Well," Zip continued slowly, "you can't see the west half of the alkylation unit. It's covered in a butane fog. The butane cloud is drifting past the plant fence, out towards the state highway. If someone drives past and flips out a lit cigarette, we might just blow away half of Texas City."

"Zip, can't you isolate the leaking line from the reactors?"

"No good, Norm," Zip sighed. "The effluent gate valves must be eaten up with acid too. I closed them all real tight, but they won't hold. The whole reactor train is depressuring butane through the leak."

A little mental math predicted that if the entire liquid content of all 10 reactors vaporized through the leak, then a rather large vapor cloud would be formed. Actually, the volume of the reactors, settlers, exchangers, and piping contained 10,000 barrels of butane. If the entire 10,000 barrels were to form a vapor cloud and ignite, the resulting detonation would approximate the force generated by the atomic bomb dropped on Hiroshima in 1945. The resulting destruction of Texas City would probably negatively impact my career at Amoco Oil.

Twenty minutes later, I was standing underneath an 8-inch line spewing a jet of liquid isobutane. I watched it flash to a white cloud in the warm Texas night. How had this happened? What had caused a sudden leak to develop on the new carbon steel line?

Figure 46.1 tells the tale. A faulty level indicator on the acid settler drum had caused the 90 percent sulfuric acid to carry over into the caustic settler drum. One pound of acid will neutralize 10 lb of the caustic solution. When I checked the pH of the circulating solution from the caustic settler, I measured a pH of less than 1. That is, the litmus paper turned red, not its normal blue.

Strong sulfuric acid above 85 percent strength is entirely noncorrosive. As the sulfuric acid is diluted with water, it becomes exponentially more corrosive to carbon steel. Small amounts of weak sulfuric acid were carrying over into the reactor effluent line from the caustic settler.

"But, Norm," Zip said, "I wasn't that worried about this acid carry-over. This has happened before. We've been running this unit for 30 years. The old 8-inch line, why, it lasted for 30 years. It never blew out like this. Just 'cause of a little acid carry-over. Norm, what are we going to do now?"

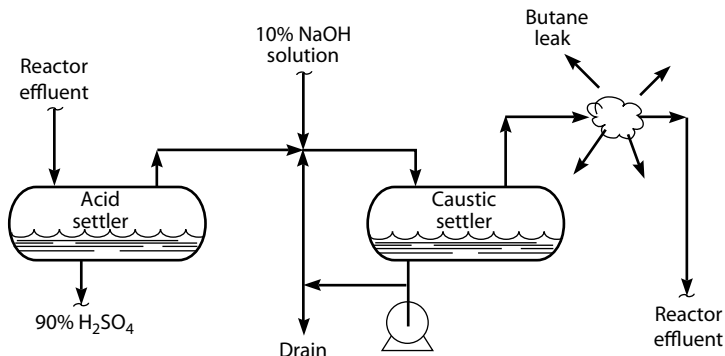


FIGURE 46.1 Piping leak due to weak sulfuric acid attack.

"Zip, the old lines were all stress relieved" (post-weld heat-treated), I explained. "All the old piping spool pieces had been x-rayed. The new piping was welded just last week here in the plant. None of it was stress relieved and the integrity of the welds . . . Well, who knows? It is the heat-affected zones of the welds that are most subject to corrosive attack, especially on non-stress-relieved carbon steel piping. The lattice imperfections in the structure of the steel caused by melting (during welding), followed by rapid cooling, causes hard spots to develop around the welds. Zip, these hard spots then create sites for the . . ."

"Norm," Zip screamed. "Shut up. Stop making a speech. What the hell are we going to do now?"

"Don't worry, Zip," I replied. "Man proposes, but God disposes." And suddenly divine intervention came to our rescue in the form of Ron Stover. Ron had returned from the storehouse with a pair of insulated gloves, a screwdriver, and a handful of brass wood screws. Ron climbed up to the leak, stuck an appropriately sized screw into the blowing hole, and twisted the screw in. The leak stopped!

Welds often represent the weak point of both process piping, and more so process vessels. It is not actually the weld metal that is the problem. It is the area on either side of the weld. The steel was melted during welding and then refroze. The next true story is about a failure of a weld on a process vessel with a rather tragic ending. The names have not been changed to protect the guilty.

46.6.2 Hydrogen-Assisted Stress Corrosion Cracking

July 17, 1983, was a day that will live in infamy in the petroleum refining industry. On that day, 16 men and 1 woman lost their lives at the Unocal (now Citgo) Chicago Refinery (actually located in Lemont, Illinois). Months later, I was hired to investigate the cause of this terrible incident.

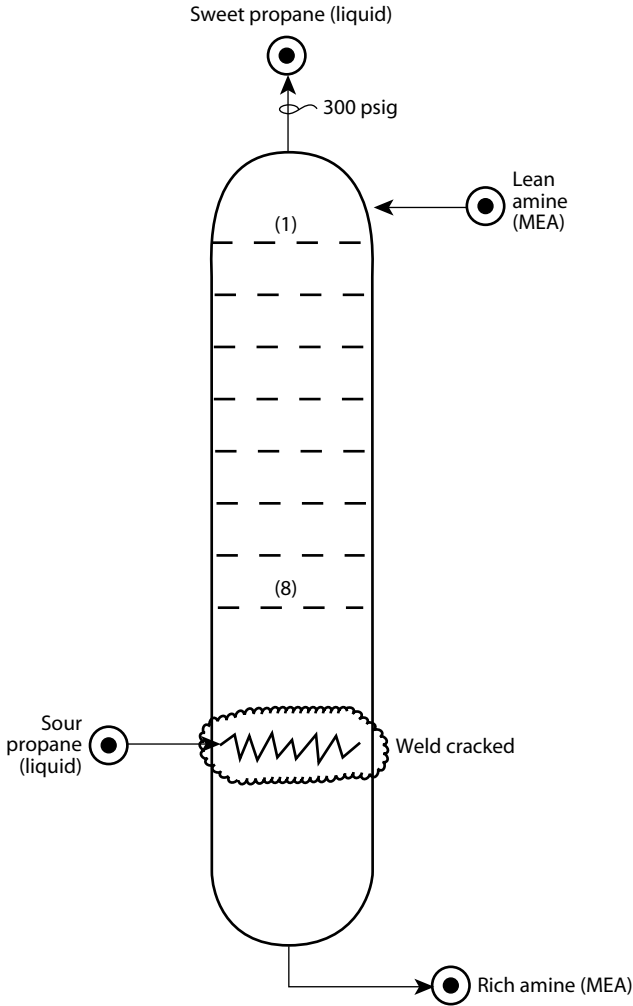


FIGURE 46.2 Hydrogen-assisted stress corrosion crack causes vessel to fail.

A liquid-liquid extraction tower broke in half. The detonation of the resulting cloud of hydrocarbons killed most of the victims. Figure 46.2 shows the process: the extraction of hydrogen sulfide by MEA (mono-ethanol-amine). A liquid propane stream containing 3 percent hydrogen sulfide was being sweetened by contact with the MEA. Sweetening means the removal of an acidic component (hydrogen sulfide) by contact with an alkaline solvent (MEA).

The circulating MEA is a solution of 25 percent MEA and 75 percent water. The solution is not corrosive in a carbon steel vessel. I have seen such vessels used for 40 years without excessive loss of

wall thickness. Indeed, the Unocal vessel that failed after 15 years of service had not experienced any real loss of metal thickness. Yet, aqueous phase corrosion of steel exposed to wet hydrogen sulfide was the origin of the failure.

When iron reacts with hydrogen sulfide the following reaction takes place:



The iron sulfide will eventually coat the wall of the vessel to form a protective layer and retard further corrosion, as previously described in this chapter.

The hydrogen ion is just a proton. Most of these protons react at the vessel wall to form molecular hydrogen:



The molecular hydrogen is dissolved in the propane and is of no further interest to us. If the formation of molecular hydrogen from the hydrogen ions at the vessel is retarded, however, then the hydrogen ions or protons dissolve into the steel walls of the vessel. That is, protons are soluble in steel, in the same sense that salt is soluble in water.

The protons will usually pass through the vessel wall, emerge as hydrogen and float away. But if the protons encounter an imperfection in the lattice structure of the steel, they will recombine at this imperfection into molecular hydrogen. The hydrogen so formed is then trapped inside the vessel wall.

The hydrogen molecules accumulate at the imperfection in the vessel wall. The pressure builds at this imperfection. Eventually, something has to relieve this pressure. Something has to give.

If the steel is ductile, the vessel wall will bulge internally and a *hydrogen blister* will form. It looks rather like a piece of plywood that has been soaked in water too long.

If the steel is hard, because it has lost its ductility, the vessel wall will crack. The crack will propagate along the hardened zone. The hardened area is created when the heat-affected zone of a weld cools too rapidly without being post-welded (i.e., stress relieved) heat treated.

Cyanide salts are known to prevent the recombination of protons to molecular hydrogen at the vessel wall. These cyanides are supposed to be removed from the tower's feed by extraction with thiosulfides in the upstream washwater. But Unocal was not adding any thiosulfides to their washwater.

Acetic acids are known to promote hydrogen ion penetration of vessel walls. Acetic acid is formed from the thermal decomposition of heat stable salts formed due to oxidation of the MEA. These heat stable salt precursors were supposed to be removed in an MEA reclaimer. But Unocal had not used the reclaimer unit for many months.

The cyanides and the acetic acid by themselves would not likely cause the vessel wall to crack. It was the concentration of hydrogen

sulfide in the MEA solution that may have been the main culprit. The moles of hydrogen sulfide divided by the moles of monoethanol-amine is called the *amine molal load*. This amine load should be kept below 0.45. The amine load at the Unocal plant the day before the explosion was not accurately known. But it was definitely above 0.60 moles of hydrogen sulfide per mole of MEA.

Still, the vessel would not have failed if the vessel welds had been stress relieved by post-weld heat treatment. But this had not been done either.

Toward the end of the day, unit operators noted a small propane leak escaping from the circumferential weld (see Fig. 46.2). They sounded an alarm, and the fire crew, 16 men and 1 woman, turned out to stand by. Meanwhile the operators observed the crack spreading around the vessel at the rate of a few inches per minute.

Within moments the vessel split in half. The upper portion was later found in a cornfield half a mile away. The propane cloud released drifted across to the fire truck and was ignited by the running motor. The ensuing detonation destroyed a good part of the refinery.

Of course, the cracked weld was not a result of corrosion that day, or that week, or that month. The crack was a result of hydrogen activity and pressure accumulation inside the vessel wall for the past decade. It was a result of 10 years of proton penetration of the vessel wall. However, the final effect appeared suddenly. That's the nature of hydrogen-assisted stress corrosion cracking.

46.6.3 Erosion-Corrosion Failure

Not very far from my home in Louisiana, Shell Oil (now Motiva) operates a large refinery. In 1988, an explosion occurred in this plant. A cloud of propane vapor from the overhead of a depropanizer detonated with tremendous force. The resulting shock wave blew a 14-ft-diameter tower off its foundation. The 100-ft-high tower landed on the control room. All six operators therein were killed.

One hundred and seventy lawyers from the surrounding parishes signed up 20,000 claimants to sue Shell Oil for damages due to gross negligence. The lawyers organized the "Plaintiffs Legal Committee," which hired me to investigate the cause of the accident. Figure 46.3 shows the point of failure.

About 15 years before the explosion, corrosion of the condenser tubes had caused a series of exchanger tube leaks. The cause of failure was thought by Shell to be due to acidic components (HCN, CO₂, H₂S) in the vapor phase. To combat the tube failures, two changes were made:

- The carbon steel tubes in the condenser bundle were changed to 304 stainless steel.
- A neutralizing ammoniated water injection stream was introduced through an injection quill, as shown in Fig. 46.3.

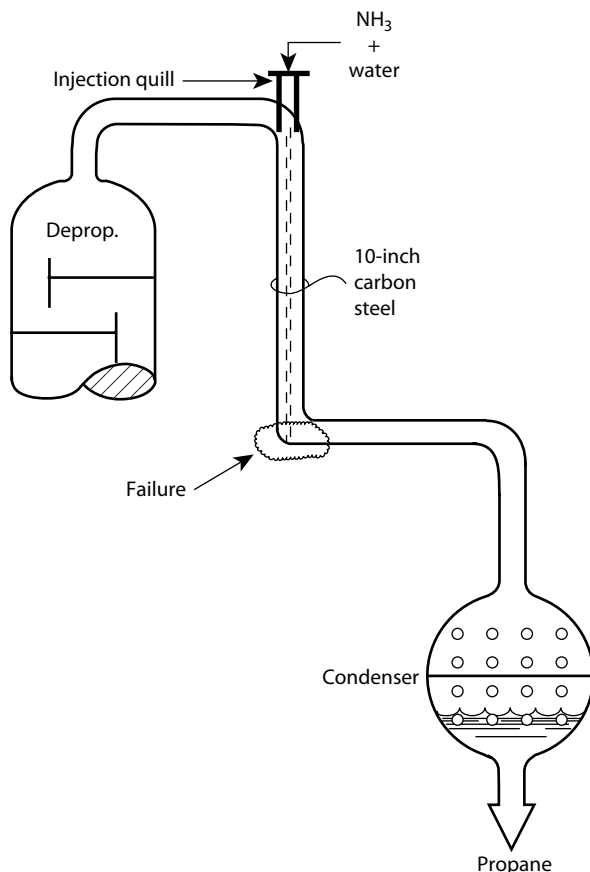


FIGURE 46.3 Impingement of water on an elbow causes failure.

Question: If the carbon steel condenser tubes were failing due to acid attack, how about the carbon steel piping? Okay, maybe the piping upstream of the condenser was dry. After all, acidic vapor phase components in a water-free environment are not corrosive. No water was condensing upstream of the condenser. But once the waterwash injection was started, the carbon steel piping upstream of the condenser was wet.

But, dear reader, certainly the ammonia in the injection water would neutralize the acidic components as they dissolved in the water and rendered the water noncorrosive? This is both true *and* false.

The water was not injected through a spray nozzle into the pipe. Hence, the washwater was not dispersed. Only the periphery of the water was contacting the vapor phase. The major portion of the washwater remained basic due to the excess of ammonia dissolved in the water.

Only the peripheral area of the injected water came into intimate contact with the acidic vapor phase in the pipe. Hence, only this peripheral water became acidic as the ammonia in the water was neutralized by the H_2S , CO_2 , and cyanides in the vapor.

You might think this is an interesting but unsupported theory. Not true. Referring again to Fig. 46.3, note that the point of failure was the elbow downstream of the injection quill. A section of the piping on the outer radius of the elbow failed (but the piece of steel that ripped off the piping remained partly attached to the piping). When I inspected the failed elbow, I found the pipe itself had suffered little loss in thickness. The 6 by 15-inch piece of steel ripped away from the pipe was mostly just as thick as the pipe. But the edge of this 6 by 15-inch piece of steel was paper thin. The acidic water at the periphery of the water injection stream had cut a groove in the steel, and the pipe had failed around this groove.

46.6.3.1 Errors Promoting Erosion-Corrosion

The most serious error made by Shell was not dispersing the water into the pipe with full-cone spray nozzles. Other errors that may have contributed to the failure are:

- The injection water had not been deaerated. That is, it was just Mississippi River water that contained dissolved oxygen. Oxygen is highly corrosive to carbon steel.
- The water was injected in the direction of flow of the vapor. Also, the water was injected at the top of a vertical run of pipe. Both of these effects accelerated the velocity of water as it struck the vulnerable elbow. This promoted erosive forces.
- Nothing had been done to extract the hydrogen cyanides from the vapor. These cyanides are dangerous in that they promote hydrogen ion penetration of steel. They are best extracted from depropanizer feed by the use of thiosulfides upstream of the depropanizer. The thiosulfides should be dissolved in washwater, which will react with the HCN (but not the CO_2 or H_2S), and used to wash the depropanizer feed.

46.6.3.2 The Settlement

The case was settled out of court for \$170 million. The seven principal lawyers on the Plaintiffs Legal Committee each cleared \$8 million. I suspect that Shell Oil settled partly because they did not want to address the following question: If the carbon steel condenser tubes were failing due to corrosion, what was preventing the carbon steel piping upstream of the condenser from failure? It's true that the piping was 0.25 inch thick, and the wall thickness of the condenser tubes was 0.1 inch. But this only means the piping would take longer to fail than the tubes.

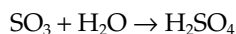
46.6.4 Summary

Combating corrosive failures requires an understanding of chemistry, fluid flow, vapor-liquid equilibrium, and thermodynamics. Any time water, air, and acids get together and impact steel at a high velocity, rapid rates of metal loss can be anticipated. If cyanides are present, metal failure may be greatly accelerated. Failure rates are accelerated at locations where the crystal lattice structure of the steel has been distorted by welds which have rapidly cooled and then not reheated to remove stresses.

46.7 Corrosion—Fired Heaters

Sulfur is a common contaminate of fuel. For example, pipeline-quality natural gas may contain 20 ppm of hydrogen sulfide and mercaptans. Clean refinery fuel gas typically has 100 to 200 ppm of total sulfur. Number 2 furnace oil (at least in the United States) will have perhaps 1000 ppm of sulfur. Heavy industrial fuel oil (No. 6 fuel oil or bunker fuel oil) may contain 6 percent (60,000 ppm) sulfur. About 90 percent of the sulfur in these fuels is converted to sulfur dioxide. However, about 10 percent is converted to sulfur trioxide (SO₃).

As the flue gas cools, it will react with the water formed from the combustion of hydrogen in the fuel to produce sulfuric acid (H₂SO₄):



The sulfuric acid may precipitate out of the flue gas as follows:

- 250°F, if the fuel had 100 ppm of sulfur
- 400° to 450°F, if the fuel had 6 percent (60,000 ppm) sulfur

Don't take these temperatures too literally. In practice, the precipitation temperatures for sulfuric acid are higher by 50°F plus. The reason is that not all of the flue gas from the convective section of a heater is at a uniform temperature.

If the average flue-gas temperature in the stack is 300°F, then some of the flue gas will be cooler and some will be hotter. The cooler gas will then begin to precipitate sulfuric acid. This acid reacts with the convective section finned tubes to form iron sulfate. A thick, sticky, moist, acidic, grayish deposit will begin to plug the convective section tube bank. The flow of flue gas will be restricted. A reduced draft and a positive pressure will begin to develop in the firebox.

46.7.1 Stack Integrity

As I write this, I have just returned from a client's plant. The stack on one of their heaters reminded me of the leaning tower of Pisa in Italy. The fuel gas to this heater sometimes has up to 10 percent hydrogen sulfide, with a stack temperature as low as 500°F. Sulfuric acid attacked the base of the stack and undermined its structural integrity.

Thus, disregarding environmental constraints, there are good reasons to limit the sulfur content of fuel gas.

46.7.2 Air Leaks

Air leaks in the convective section of a heater will reduce the concentration of SO_3 . However, this small beneficial dilution is of little importance compared to the cooling effects of the tramp air. Air leaks in any portion of a heater's convective section or stack will promote localized precipitation of sulfuric acid, with resulting corrosion and fouling.

On the other hand, anything (even tramp air leaks) that increases excess air in the firebox will increase the convective section flue-gas temperature and dilute the SO_3 in the flue gas. This will suppress sulfuric acid condensation and corrosion, but at the cost of reducing furnace fuel efficiency.

46.7.3 Air Preheaters

A typical fired heater combustion air preheater is shown in Fig. 46.4. Combustion air at a pressure of 5 inches of water is shown on the tube side. The flue gas is on the shell side, flowing at a draft of 2 inches of water. This air preheater is clearly leaking cold air into the flue gas. How do I know this?

The mass flow of flue gas is typically 10 percent greater than the mass flow of combustion air. Without air leaks, but with some ambient heat losses, the following should be approximately true:

- ΔT of flue gas equals ΔT of combustion air.

In Fig. 46.4, the temperature rise of the air is 300°F, while the temperature drop of the flue gas is 500°F. The cold combustion air is quenching the hotter flue gas. About 50 percent of the air is leaking through the corroded tubes.

The cause of the tube leaks is cold flue gas. But the cause of the cold flue gas is tube leaks. The real cause of the tube leaks is sulfuric acid corrosion. The sulfuric acid condensation is the result of the low temperatures of the flue gas. To avoid this problem, the flue gas must not be cooled too much by the cold air.

Many air preheaters will have a steam preheat section to heat the combustion air by 60 to 80°F. While this may reduce the energy savings of the air preheater by 25 percent or more, my experience is that it's an effective way to reduce air preheater tube leaks. This is especially true if the fuel has a few percent sulfur.

46.7.4 Plugged Air Preheaters

Cold air leaks promote exterior corrosion of the tubes shown in Fig. 46.4. The resulting corrosion deposits restrict the flow of flue gas and causes a positive pressure to develop in the firebox. This forces the operators to reduce air flow, which then reduces the temperature

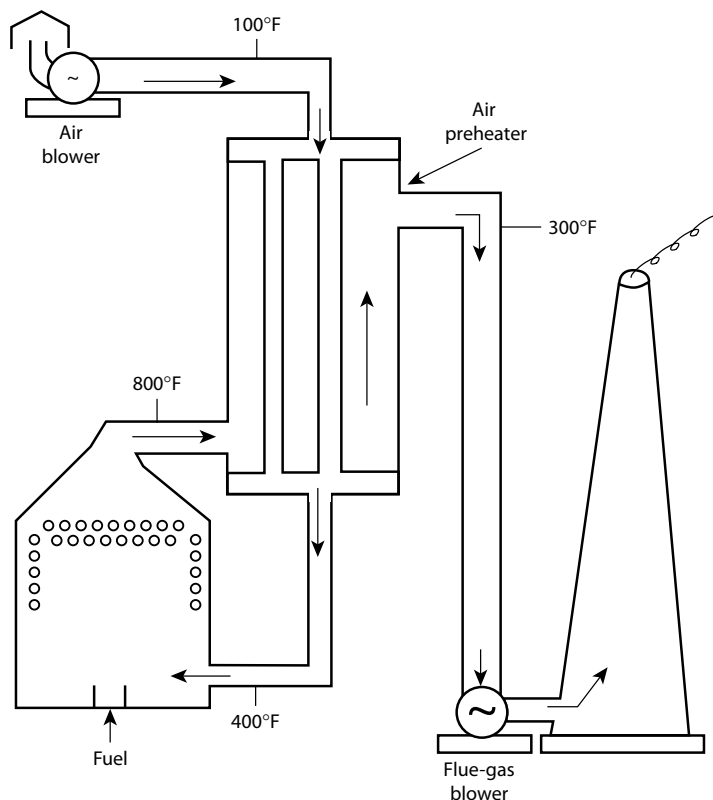


FIGURE 46.4 A leaking air preheater.

in the flue gas further. Also, the concentration of SO_3 in the flue gas rises. Sulfuric acid attack is accelerated.

I have seen air preheaters with the flue-gas-side ΔP increase by 300 percent in just two years. This same preheater was leaking so badly that 60 percent of the combustion air was leaking directly up the stack through the preheater and thus bypassing the firebox and burners.

46.7.5 Reducing Air Preheater Failures

An air-side bypass around the preheater can be used to avoid overcooling the flue gas. Air preheat with steam upstream of the preheater is the most effective method to suppress leaks.

Ambient heat losses on the flue-gas side can be an important factor in promoting sulfuric acid condensation on smaller units. Ambient heat loss also decreases heater draft due to increased flue-gas density in the stack.

46.8 Oil-Fired Heaters

In the radiant section of heaters or boilers, it's not sulfur that causes the problem, but vanadium. All black residual fuel oils contain vanadium, along with some amount of sodium. It's the sodium that causes the problem. The sodium forms a eutectic mixture with the vanadium.

When two metals mix together, such as sodium and vanadium, the melting point of the mixture is reduced below the melting point temperature of the individual metals. This mixture is called a *eutectic*.

Residual fuel oils may contain 1000 ppm of vanadium and 10 ppm of sodium. When the fuel is combusted, the vanadium will deposit on the exterior of the radiant section tubes. The deposit is about 0.125 inch thick. It is a hard, but brittle scale that sloughs off easily from the radiant section tubes. It does no harm. Rather, the vanadium deposits protect the tubes from overheating, and flame impingement.

Unfortunately, at about 1250 to 1300°F the eutectic of 99 percent vanadium plus 1 percent sodium melts into a viscous, sticky liquid. At any higher temperature, the mixture will run off a tube without harm. At a lower temperature, it forms the thin protective solid I described. At 1250 to 1300°F, a thick, uneven, and corrosive mixture (perhaps 1 inch in thickness) slowly eats away at localized areas of a furnace tube. The result is extremely localized tube thinning. I was mystified when I first saw such a tube in Aruba, until the local corrosion engineer explained the failure mechanism to me.

There are several things that can be done to combat this problem. Most effective is to avoid purchase of fuel oils with high sodium contents. While vanadium in fuel oil cannot normally be controlled by the refiner (i.e., crude oil will contain an intrinsic amount of vanadium), the sodium content of fuel can be minimized by the refinery producing the residual oil.

An additive can be purchased from several suppliers (Nalco, Betz) that will prevent the vanadium scale from forming. While effective, the additive is expensive.

Perhaps the best and cheapest solution is to avoid flame impingement on radiant tubes. The 1250 to 1300°F temperature needed to melt the eutectic of vanadium-sodium is above the normal tube temperature in most heaters. However, flame impingement can easily raise the exterior surface of a radiant tube above 1250°F.

Avoiding direct flame impingement does seem, in my experience, to help avoid the local corrosion associated with the vanadium-sodium eutectic mixtures.

46.9 Finned-Tube Corrosion

Apparent corrosion of the fins in a convective section, whether gas- or oil-fired, is usually not due to sulfuric acid attack, but to oxidation. If the corrosion is due to sulfuric acid, a gray, sticky solid will be present.

But if the fins have just become brittle enough to break off by hand, the problem is afterburn. The fins have simply been burned up by secondary ignition. While oxidation is a form of corrosion, we have discussed this problem in our chapter concerning the flue-gas side of fired heaters. (See Chap. 29.)

46.10 Field Identification of Piping Metallurgy

Without question, the most common single cause of fires in petroleum refineries is line failures due to excessive corrosion rates of piping of the wrong metallurgy. Usually, the error occurs during a turnaround or revamp. A piping section is replaced and one piping spool piece is fabricated from carbon steel when it should have been made of a high-chrome steel. Designed for 20 years of service, the piping then fails after just a single year.

Usually, the spool piece that failed was covered with insulation and not visible. If an operator or engineer had been alert during the turnaround, the improper piping section could have been replaced.

We, as process personnel, are not metallurgists. Still, we should be prepared to discriminate between three types of materials:

- Carbon steel: Looks slightly rusty. Is magnetic.
- Chrome steel (400 series): Looks slightly less rusty with a gray tinge. Is magnetic.
- Stainless steel (300 series): Looks more shiny than chrome steel. Is not magnetic.

The 300 series stainless steels contain nickel (316 and 317 also moly). The 400 series chrome steels are intended mainly to resist sulfur and high temperature. The 300 series steels have a greater range of chemical resistance. A handheld instrument is now available to discriminate between carbon steel, chrome steel, and different types of stainless steels.

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CHAPTER 47

Waste Water Strippers

In 1965, my first assignment as a young process engineer for the American Oil Company was to design a waste water stripper to remove phenols, NH_3 , and H_2S from refinery sour water. In 2012, somewhat older, I was troubleshooting two such sour water strippers in India. Neither of the strippers was working properly. Both were allowing excessive NH_3 to be left in the stripped sour water and developing low stripping tray efficiency.

There are three ways to design refinery sour water strippers:

- The best way
- The wrong way
- The bad way

The two sour water strippers that I was troubleshooting in India both conformed to the bad method of process design.

47.1 Purpose of Sour Water Strippers

Refinery sour water originates largely from delayed coker, hydrodesulfurizer reactor effluents, and catalytic cracker and visbreaker fractionators. The main contaminants are NH_3 and H_2S . Sour water stripper bottoms are reused in two places:

- Washwater for the crude unit desalter on a once-through basis to remove chloride salts that promote hydrochloric acid corrosion in the crude tower overhead condensers
- Makeup washwater for hydrotreater effluent recycle washwater to remove ammonia sulfide salts that plug downstream condensers and heat exchangers

When sour water stripper bottoms are used in the crude unit desalter, the NH_3 should be about 10 to 20 ppm. Higher NH_3 levels

interfere with crude unit corrosion control programs due to chlorides. I've explained this in greater detail in my book, *Troubleshooting Process Operations*, 4th ed. (PennWell). One hundred ppm of NH_3 is excessive for desalter operation.

When sour water stripper bottoms are used as a makeup to hydrotreater effluent washwater, an NH_3 content of a few hundred ppm is fine. After all, the recirculated washwater has over 10,000 ppm of NH_3 anyway. Therefore, the NH_3 content of the makeup washwater is not critical, as long as 90+ percent of the NH_3 in the sour water stripper tower is stripped out.

47.1.1 The Bad Design

Figure 47.1 shows the conventional design used at most locations that I have visited in the past few decades. The operating conditions shown are typical of what I have observed recently at several refineries. The stripper feed is heated by feed/effluent exchanger with the stripper bottoms in exchanger E-1 (see Fig. 47.1). The reboiler duty (E-3) is required for the following three purposes:

- Heat the feed from 180°F to the 250°F tower bottoms temperature.
- Generate internal reflux at tray #32.
- Break the chemical bonds between water, NH_3 and H_2S , which are all endothermic reactions.

The reflux is generated in the E-2 circulating cooler. The reboiler duty is adjusted so as to control the moisture content (i.e., temperature) of the ammonia-rich gas flowing to the sulfur recovery plant thermal reactor. Too high a temperature will cause too much moisture in the sulfur plant feed. Too cool a temperature will cause NH_3 salts to plug the vapor feed line to the sulfur plant.

This design never made much sense to me. Why preheat the feed onto tray 32 with E-1, and then remove this heat in E-2? Wouldn't it have the same effect on the stripping trays to:

- Bring in the feed colder onto tray 32 (Fig. 47.1).
- Reduce the heat extracted in E-2, the circulating cooler, to offset the effect of the colder feed.

I suppose one could argue as to the effect on stripping efficiency for trays 30–32. However, a single-field measurement is worth a thousand computer simulation calculations. I've tried shutting off the circulating reflux (E-2) and bypassing the feed preheat exchanger (E-1) to maintain a constant reboiler duty and tower top temperature. The effect that I observed on the NH_3 in the stripper bottoms was quite negligible. And why should it change? Will not the stripping factor (ratio of vapor to liquid) for the stripping trays remain unaltered?

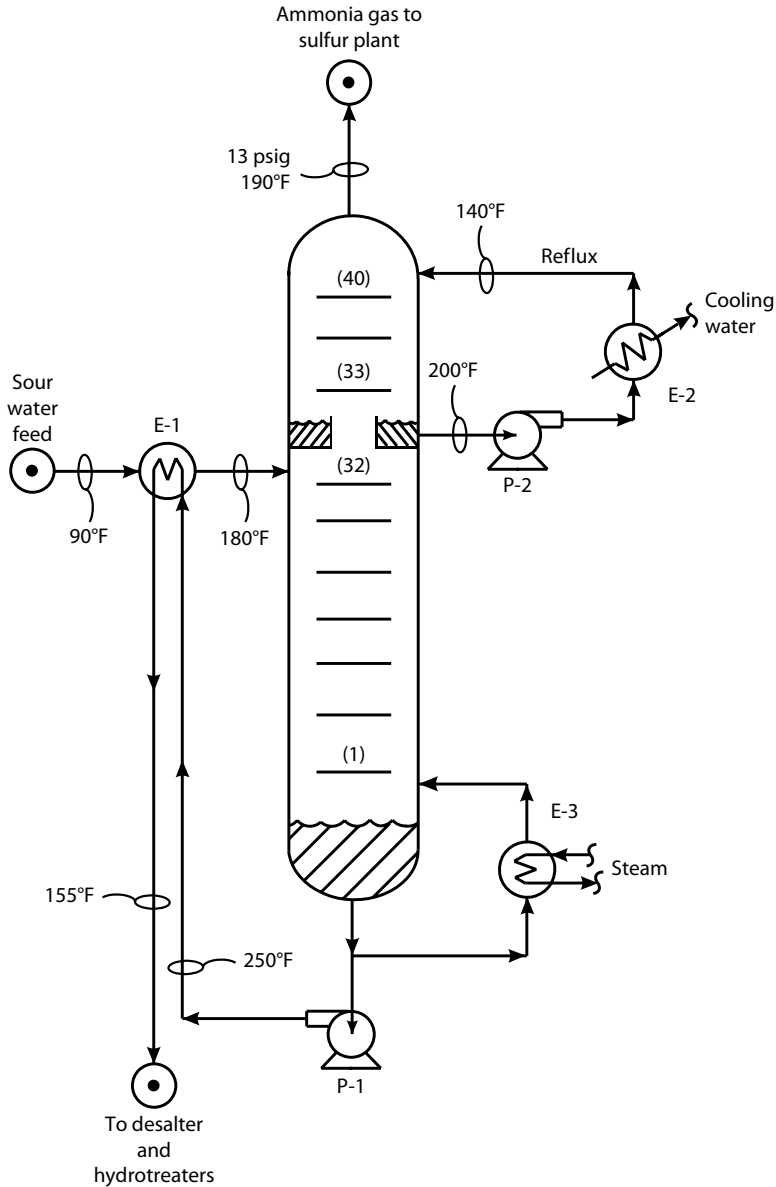


FIGURE 47.1 Conventional sour water stripper design used in newer units.

My conclusion is that the preheat exchanger (E-1), the circulating reflux loop (E-2), and trays 33–40 serve no real chemical engineering purpose. But don't take my word for this. Try it on your own unit, as I did in Mumbai, India last week.

47.1.2 The Wrong Design

Figure 47.2 is really just as wrong, but not quite as wasteful a design as shown in Fig. 47.1. At least in this version, there are only 16 trays rather than 40. My field observations indicate that a dozen or so stripping trays are sufficient to remove over 99 percent of the NH_3 from sour water and an even greater percentage of the H_2S . Note that reducing the reflux drum temperature much below 165 to 170°F will result in salt plugging of the off-gas vapor line. The 190°F tower top temperature is likely a bit conservative and will result in increased

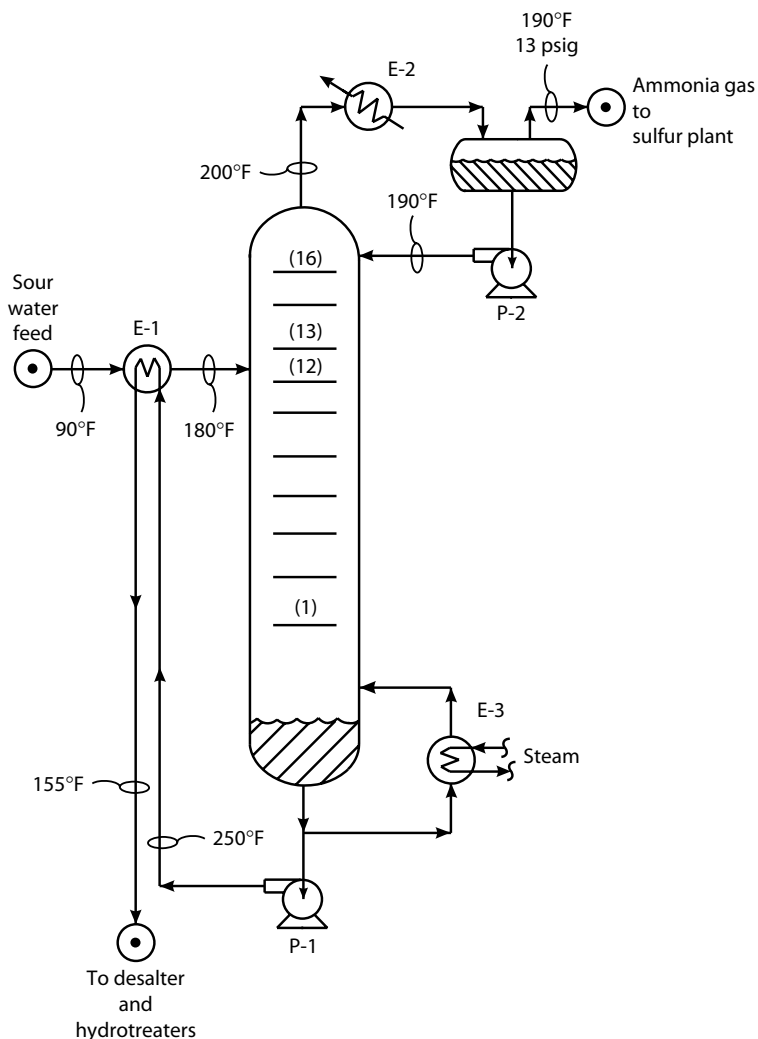


FIGURE 47.2 Conventional sour water stripper design used in older units.

water in the sulfur recovery plant feed and thus more condensation in the sulfur plant feed NH_3 gas K.O. drum. Note that this water has to be recycled back to the sour water stripper for reprocessing.

The design shown in Fig. 47.2 was common in my youth (i.e., the 1960s). However, it also suffers from the same heat balance drawbacks and needless complications that I discussed for Fig 47.1. That is, a lot of equipment is added to generate reflux. This is very wasteful, as there is no fractionation required between the stripper feed and the overhead product. All we are trying to do is strip out the NH_3 and H_2S . That's why it's called a sour water STRIPPER!

In 1969, while working for the Amoco International Oil Company in the United Kingdom, I designed a sour water stripper that eliminated the unnecessary features of the unit shown in Fig. 47.2, which worked just fine.

47.1.3 The Best Design

Figure 47.3 shows the essentials of this 1969 design. Feed is brought in at essentially ambient conditions (70°F–100°F) from the sour water feed tank. To heat the feed from 90 to 250°F requires about 16 wt.% steam flow, or about 1.3 to 1.4 lb of steam per gallon of stripper bottoms. This value is close to a typical design stripping steam ratio for sour water strippers. The E-1 feed preheater, reflux pump (P-2), and reflux cooler (E-2) of Fig. 47.2 are all eliminated. How, then, do I know that the design shown in Fig. 47.3 will work? Because it was built this way (at Milford Haven in Wales) in 1970, and it worked as well as the conventional design shown in Fig. 47.2.

But why hasn't this simple process design configuration been adopted by the industry in the last 40 years? For those of you who have read my other books, you will find that the answer is that there is no answer.

47.2 Two-Stage Sour Water Stripper

Figure 47.4 shows a sour water stripper with a side draw-off. Partly stripped water is extracted from tray #8 and is directed to the hydrotreaters for use as makeup water in the ammonia salt removal step of the desulfurizer reactor effluent. Completely stripped water from the sour water stripper bottoms is sent to the crude unit desalter. I designed such a modification for the former Coastal Corporation in Aruba, but I don't know if it was ever installed. However, this is just a direct application of the "Lean, Semi-Lean Amine Regeneration Design" that is often used in the natural gas industry to save energy (see my book, *Troubleshooting Natural Gas Processing*). I have designed such two-stage rich amine regenerators, which is quite conventional technology, with the objective of producing a very low residual H_2S content in the lean amine solvent, at the expense of a higher residual H_2S content in the semilean amine solvent.

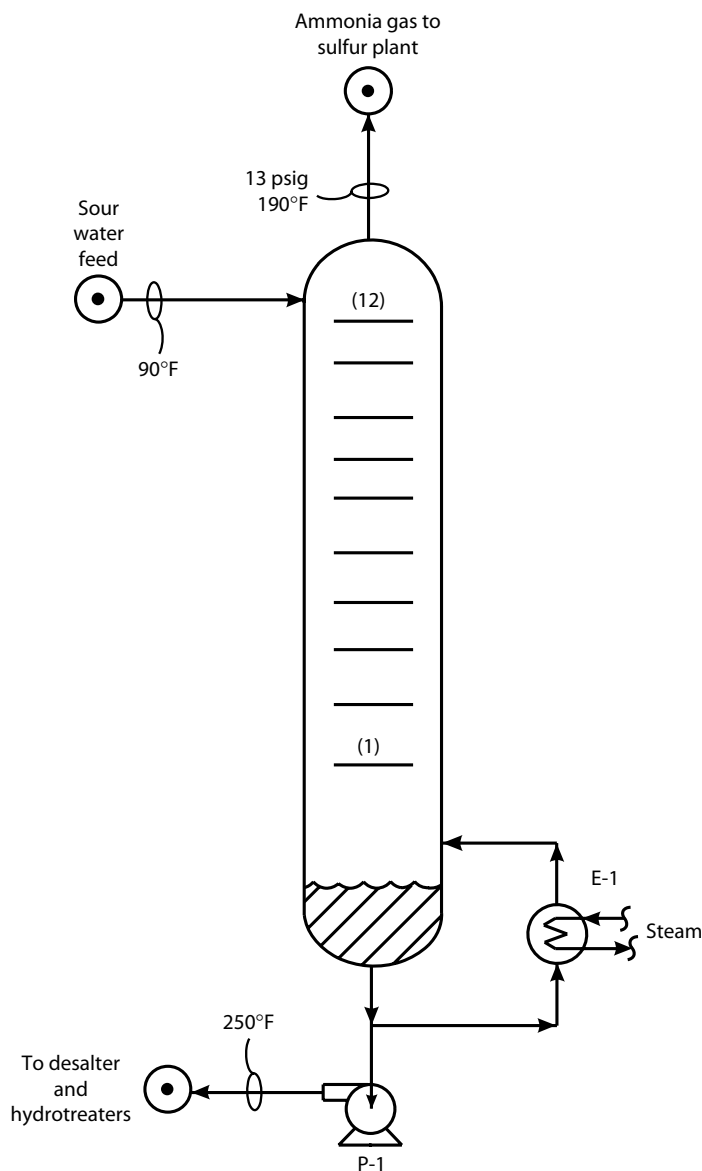


FIGURE 47.3 Correct sour water stripper design without feed preheat or external heat extraction.

47.3 Tray Efficiency

For one of the sour water strippers I worked on in India, the NH_3 in the bottoms product was about 200 ppm, even though the NH_3 in the feed was only 1000–2000 ppm. My client felt, and correctly so, that

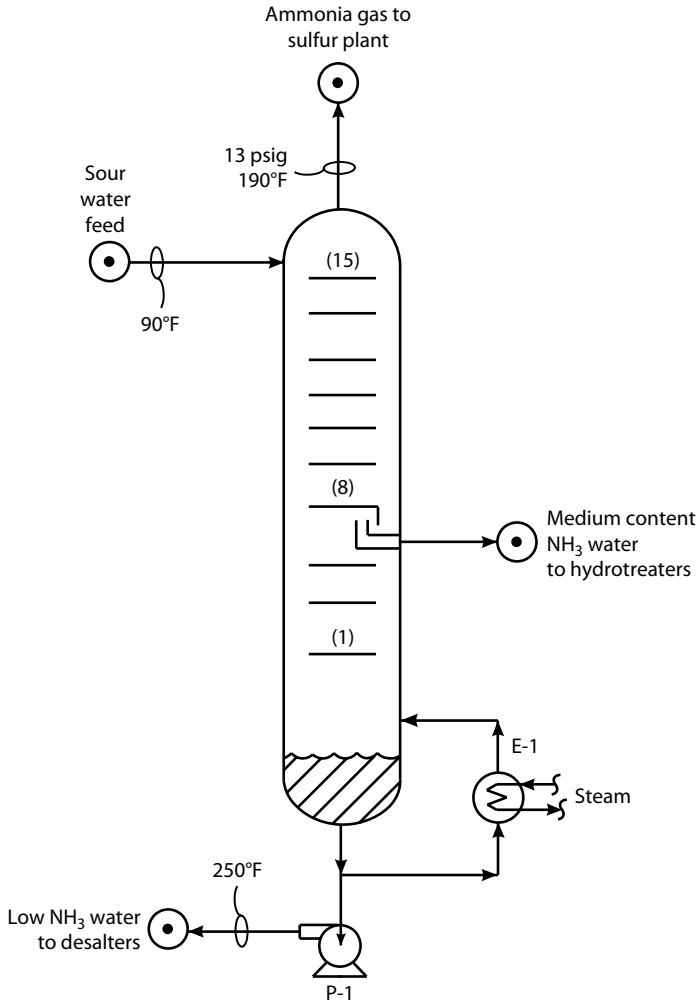


FIGURE 47.4 Two-stage sour water stripper design without feed preheat.

the 200 ppm of NH_3 , precluded the use of this sour water stripper bottoms as a crude unit desalter washwater. Thus, the water flowed directly into the refinery effluent treatment plant.

The problem was not the lack of stripping steam. The problem was also not the excessive pH of the stripper bottoms as my client in India suspected. It is quite normal for the stripper bottoms to have a higher pH than the feed, if the stripping efficiency is inadequate. That is, the acidic H_2S is stripped out of the sour water more easily than the basic ammonia. The problem of excessive NH_3 in the stripped water effluent was low tray efficiency.

Climbing up this tall tower with far too many trays (see Fig. 47.1) (I'm more aware of such oversized towers at age 70 than I used to be), I observed:

- The delta P across the bottom 32 trays was zero. Meaning there were no intact trays.
- The delta P across the top 8 trays was 7 psig (or about 16 ft of water—which equaled the vertical height of the tower above the chimney tray shown in Fig. 47.1. Meaning that trays 33–40 were totally flooded.

I advised my client as follows:

- Do not use valve or flutter cap trays. The valve caps stick to the tray deck and promote high-pressure drop and flooding. Use a grid tray with a one-half-inch cap lift. Pro-valves are a good option.
- Bolt the trays to the tray ring, or use back-to-back trays with shear clips (see my book, *Process Design for Reliable Operations*, for mechanical details). This will help to avoid tray failures due to pressure surges caused by high liquid levels.

In the Coastal Refinery in Aruba, we used sieve trays with one-half-inch holes, which seemed to work fine. Avoid packed towers. They are subject to vapor-liquid channeling and poor fractionation efficiency due to sloppy installation, fouling, or poor liquid feed distribution. When calculating the required hole area for the trays, don't forget that the vapor loads and hence the required tray hole area will substantially diminish as the vapor flows up the column from the reboiler to the feed tray. This will normally require a reduction in the tray deck hole area in proportion to the reduced vapor flow rate.

47.4 Computer Simulation and Theoretical Tray Efficiency

A rather detailed and complete article on sour water stripper tray efficiency was published in 2012.¹ It compares three ways of looking at tray efficiency for stripping NH_3 out of water (reference 1):

- Murphree vapor efficiency
- Overall tray efficiency
- Mass transfer tray efficiency

In general, according to this reference, stripping efficiency for NH_3 is 40 percent. For H_2S it's quite variable. However, H_2S is so easily stripped out of water, compared to NH_3 , that this variability for H_2S stripping is not important.

The problem with this sort of approach to tray efficiency is that it is largely irrelevant. The main problems we have with tray efficiency in sour water strippers are:

- The trays are not level.
- The weirs are not level.
- The valve caps are sticking to the tray decks or blown off.
- The trays are corroded through.
- The tray decks have been blown out due to misoperations.
- Trays are leaking because the deck hole area is excessive for a given vapor rate.

Of course, no one could take into account such factors. They are unknown. But what is known is the performance of existing sour water strippers in pretty much identical services. There are decades worth of operating data available for sour water strippers. I helped design a sour water stripper in 1965 (with dumped ceramic packing, which did not work) for American Oil. It's far safer to design a new sour water stripper based on a prototype than on a purely theoretical basis.

However, Reference 1 does contain a useful analysis of a factor that I have often thought about. That is, the trade-off between steam consumption and the number of trays. From Fig. IV of the reference, note that increasing the number of stripping trays from 18 to 32 reduces reboiler steam consumption by 23 percent for the same NH_3 concentration in the stripped water.

I suppose this is a good argument (from the designer's perspective) to design sour water strippers with 30 trays. However, from the unit engineer or operator's perspective, it illustrates the potential energy savings from maintaining the stripper trays in good mechanical condition and to ensure tray deck levelness. That is to say that if trays are in good mechanical condition and properly installed so that they work as intended, then one would need less reboiler steam.

47.5 Use of Caustic to Improve Stripping

I was working on a smaller crude unit in Alabama to improve desalter efficiency. Their problem was the periodic carry-over of emulsified brine. The depth of the emulsion between the crude and water phases in their desalter was excessive. The pH of the washwater from the sour water stripper was $8\frac{1}{2}$ to 9, even though the NH_3 content of the stripper bottoms was zero. Talking to the operators, I discovered that they added NaOH to the sour water stripper feed to diminish the NH_3 content of the stripped water.

Caustic contamination of the desalter washwater is known to promote high desalter vessel emulsion levels. There are dozens of references to this problem in my favorite source of technical information.

That is, the old back issues of the NPRA Q&A (i.e., The National Petroleum Refiners Associated, Question and Answer) sessions. With this authoritative source as support, I asked the operator to stop the NaOH addition to his sour water stripper feed. Two hours later, at the same reboiler duty, the NH_3 content of the stripped sour water had gone from 0 to about 5 ppm. And the emulsion layer in the crude unit desalter had been reduced enough so that brine carry-over was no longer a problem. The 5 ppm of ammonia was quite acceptable.

Caustic addition to the sour water stripper feed will increase NH_3 stripping efficiency by reducing the solubility of NH_3 in water. This may be a good practice if the stripper bottoms is flowing into the refinery effluent treatment basin. But I would rather have 10–20 ppm NH_3 in the sour water stripper bottoms than contaminate my desalter with NaOH, which promotes the formation of brine–crude oil emulsions.

“Why then, Mr. Lieberman, were we using caustic in our sour water stripper all these years?” asked the young operator. “That caustic’s expensive.”

“The answer, son,” I explained, “is that there is no answer.”

47.6 Water Stripper Reboiler Corrosion and Fouling

One of the questions that I am often asked is what can be done to mitigate reboiler tube failures in sour water strippers and the consequent steam leaks into the stripped sour water?

One helpful suggestion is to avoid the use of kettle reboilers or once-through thermosyphon reboilers, in favor of circulating thermosyphon reboilers. That is, feed the reboiler directly from the bottom of the stripper and design for about a 10% to 20% vaporization rate (see Fig. 47.3). This will reduce, but unfortunately not eliminate, the reboiler tube failures by maintaining more liquid flow across the tubes.

I have a simpler suggestion, if you can meet the following process limitations:

1. The stripped water flows to the crude unit desalter and/or the hydrotreater effluent washwater makeup.
2. External water is required to supplement stripped sour water to the above units. That is, the plant’s sour water stripper bottoms is not sufficient for the total washwater requirements.

My suggestion is to use open steam in the stripper bottoms and abandon the reboiler. This will increase the stripper bottoms water flow by 10% to 20%. But this will also be offset by the diminished freshwater makeup rate to the desalter or hydrotreater effluent wash.

I would estimate that half of my clients use open stripping steam and have discarded their reboilers. Or they are operating with massively leaking reboiler tubes, which is really pretty much the same thing as open stripping steam.

Referring to Fig. 47.3, for many refinery operations, the circulating thermosyphon reboiler shown can be eliminated and replaced with the same quantity of stripping steam, thus further simplifying the design of your waste water stripper installation.

For many waste water strippers used in refinery services, rapid rates of reboiler fouling are experienced, not due to corrosion, but as a consequence of heavy hydrocarbon sludge, that settles out in the bottom of the sour water stripper feed tank. In particular, slurry oil that is produced in a refinery catalytic cracking unit can accumulate in the stripper's feed tank. This slurry oil, being denser than water, settles out in the tank and eventually is drawn into the suction of the sour water stripper feed pump. The heavy hydrocarbon phase then accumulates in the stripper's reboiler, and with time and temperature fouls the reboiler's tubes.

I solved this problem at the Mobil Oil Refinery in Coryton (United Kingdom) by elevating the suction nozzle for the pump on the sour water storage tank a few feet above the tank's bottom drain. A more common solution would have been to use a floating suction inside the tank.

47.7 Ballast Water Stripper

Benzene is quite soluble in sea water. Thus, crude tankers produce ballast water with high benzene contents, which should be stripped out before the ballast water is drained to the sea. I was working with a refinery in Baytown, Texas, on such a waste water stripper. It was a new facility that was not effectively achieving its design function of rejecting the benzene to an oxidizer. I've published the details in the *Oil and Gas Journal* (July 18, 1994, pp. 51–54). The story illustrates that no matter how good the design, confused operators, supported by inexperienced engineers, can ruin the performance of a waste water stripper.

47.8 Conclusions

Generally accepted designs in the refining industry are not necessarily correct. The application of basic chemical engineering principles such as the stripping factor can often lead to simpler, more cost-effective designs. Avoid the use of flutter cap-type valve trays in fouling services. The caps will stick to the tray deck and cause flooding. The high pH of stripped sour water is a symptom of poor stripping efficiency, not the cause. Preheating sour water stripper feed normally serves no purpose. Avoid the use of caustic if the stripped water is to be reused as the crude unit desalter washwater as it will promote an emulsion. A refinery sour water stripper should have about a dozen trays, not 40. Best to use grid or sieve trays, rather than random-type packing (i.e., rings) in the tower. If high turndown rates are required, use

bubble caps rather than perforated or valve tray decks. For many locations, the bottoms reboiler may be replaced with open stripping steam. The principles that I have discussed for refinery sour water strippers also apply directly to waste water strippers in general.

Reference

1. Nathan Hatcher and Ralph Weiland, "Reliable Design of Sour Water Strippers," *PTQ* Q3, 2012, pp. 83–91.

CHAPTER 48

Fluid Flow in Pipes

Basic Ideas to Evaluate Newtonian and Non-Newtonian Flow

We have spent a lot of time in this book so far discussing the movement and processing of fluids through different types of equipment, but how is the process fluid conveyed from one piece of equipment to another?

In general the answer is, “It is carried by pipes,” and in most cases the pipes are circular cross-section and made from carbon steel.

48.1 Field Engineer’s Method for Estimating Pipe Flow

One day I asked Norman how he would calculate the pressure loss for flow through circular cross-section pipes—the sort of pipes that are used throughout most process plants that we work in. I felt sure he would have some simple method. Here is the method with an example of its use for a pressure drop survey, more or less as he gave them to me.

48.1.1 Low-Viscosity Liquids and Vapors in Turbulent Flow

The pressure drop (ΔP) for fluids flowing through pipelines of any diameter may be approximated by the following formula. I’ve checked this correlation in the field. Caution! This formula only applies if viscosity is low and flow is turbulent. By low viscosity, I mean something less than 2 or 3 cP, or perhaps 10 SSU (seconds Saybolt universal). Water, warm diesel oil, and hot gas oil all fall within this category.

Also please note that this formula assumes the pipes are reasonably smooth steel. Norman says that if he is dealing with older cast iron pipes in water service or other services that promote corrosion deposits, scaling, or surface roughness, he increases the calculated ΔP by 20 percent. If the piping has many elbows, fittings, and valves, he

increases the calculated pressure drop by 50 percent. If the piping has a reducer, he treats the reducer as if it is a nozzle.

$$DP_{100\text{ ft}} = \frac{4.4}{ID_{11}} \times \frac{D}{62.3} \times \frac{V^2}{28}$$

which simplifies to

$$\Delta P_{100\text{ ft}} = \frac{D \times V^2}{400 \times ID_{11}}$$

Caution: This equation only holds for turbulent flow when expressed in the following units. It was derived from charts used to size industrial piping systems. It is designed to provide a conservative high value for pressure drop. To verify that the flow is turbulent, refer to Sec. 48.3.3.

- $P_{100\text{ ft}}$ = Pressure drop in psi, per 100 equivalent ft of pipe
- D = Density of the flowing liquid, lb/ft³
- V = Linear velocity of the fluid, ft/s
- ID_{11} = Pipe inner diameter, inch

48.2 Field Pressure Drop Survey

An engineer has been assigned to check a kerosene process line for ΔP . The chief operator suspects the line is partly plugged (see Fig. 48.1):

1. Kinematic viscosity of kerosene is 0.6 cSt. Therefore, viscosity is not a critical factor because it is less than 5 to 10 cSt.
Note that centistokes = centipoise divided by density. The density in this case is approximately 0.9g/cm³.
2. Observed (measured) ΔP is 40 psi.

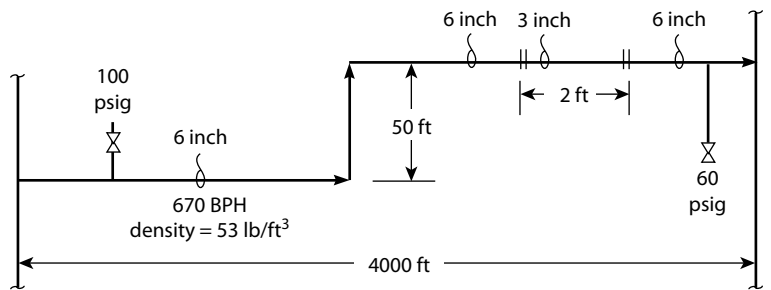


FIGURE 48.1 Pressure drop survey in a kerosene pipeline.

3. Loss of pressure due to head gain of 50 ft is:

$$\Delta P = \frac{50 \text{ ft}}{2.31} \times \frac{53 \text{ lb/ft}^3 \text{ (kero)}}{62 \text{ lb/ft}^3 \text{ (water)}} = 17 \text{ psi}$$

Note that the 2.31 factor is the height in feet of a column of water that will exert a pressure equal to 1 psi, and atmospheric pressure 14.7 psi is equivalent to a height of approximately 30 ft of water.

4. The actual flowing pressure loss corrected for elevation is then:

$$40 \text{ psi} - 17 \text{ psi} = 23 \text{ psi}$$

5. The 6-inch diameter line has a short 2-ft long restriction where the pipe size is reduced to 3-inch diameter (see Fig. 48.1). Line lengths this short (i.e., 2 ft) can be treated as a nozzle as follows:

$$\Delta P = \frac{0.34 \times 53 \text{ lb/ft}^3 \text{ (kero)}}{62 \text{ lb/ft}^3 \text{ (water)}} \times \frac{(20 \text{ ft/sec})^2}{28}$$

The 0.34 is a nozzle-loss coefficient. The 20 ft/s is the velocity of the kerosene in the 3-inch restriction. The 28 factor converts ΔP from inches of water pressure drop to psi.

The ΔP over the 3-inch restriction is 4 psi.

6. The remaining ΔP due to friction loss in the 6-inch line is then:
 $23 \text{ psi} - 4 \text{ psi} = 19 \text{ psi}.$
7. To calculate the pressure drop in the 6-inch line per 100 ft of line, first find the linear flow rate of the kerosene in ft/s.

670 barrels per hour in a pipe of approximately 6-inch inner diameter. Cross-sectional area A of pipe, ft^2 :

$$\begin{aligned} A &= \pi (\text{ID}/2)^2 \\ &= \pi [(6/12)/2]^2 \\ &= \pi 0.0625 \\ &= 0.2 \text{ ft}^2 \end{aligned}$$

Linear flow rate V , ft/s:

$$\begin{aligned} &670 \text{ BPH in 6-inch ID pipe (see Fig. 48.1)} \\ &= 670 \times 5.56 \text{ ft}^3/\text{h in 6-inch ID pipe (where } 5.56 \text{ ft}^3 = 1 \text{ barrel)} \\ &= \frac{670 \times 5.56}{60 \times 60} \text{ ft}^3/\text{s in 6-inch ID pipe} \\ &= 1.03 \text{ ft}^3/\text{s} \end{aligned}$$

Linear flow rate, $V = \frac{1.03}{0.2} \text{ ft/s} = 5 \text{ ft/s}$
Then:

$$\Delta P = \frac{53 \text{ lb/ft}^3(\text{kero}) \times (5 \text{ ft/s})^2}{400 \times 6 \text{ inch}}$$

The 6-inch approximates to the actual pipe inner diameter of 6.06 inch, for a nominal pipe diameter of 6 in. The 400 is a coefficient so that the ΔP in psi is expressed as ΔP per 100 ft of line. The ΔP per 100 ft of line is then 0.5 psi.

8. For 4000 ft of pipeline, the ΔP is found from:

$$\Delta P \text{ (due to friction)} = 0.5 \times 4000/100 = 20 \text{ psi}$$

9. There are only two elbows in this line. Hence I have neglected losses due to fittings in the long (4000-ft) line. This is not usually the case, in fact frictional loss due to fittings can sometimes double the frictional loss for straight run pipe. For further information on estimating friction losses due to fittings, consult Crane Co. Technical Paper No. 410 or your *Cameron Data Book*.¹⁻³

However, just to give you an idea of the magnitude of the frictional losses due to fittings, please refer to Fig. 48.2. We have



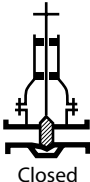
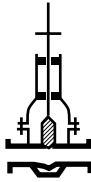
Type of Fitting		Equivalent length of straight pipe as number of pipe diameters for fluid in turbulent flow
	90° elbow (standard radius)	30 – 40
	Entry from leg of T piece Entry into leg of T piece	60 90
 Closed	Gate valve fully open Gate valve 3/4 open Gate valve 1/2 open Gate valve 1/4 open	7 40 200 800
	Fully open	

FIGURE 48.2 Frictional losses due to fittings for fluid in turbulent flow.

picked out some typical fittings and shown the equivalent length of straight pipe, in terms of pipe diameters, needed to produce the same frictional loss as the fitting for fluids in turbulent flow.

Conclusion: The ΔP due to friction (20 psi), is about the same as the corrected (step 6) ΔP of 19 psi. Therefore, there is no pluggage in the kerosene pipeline and everything is quite normal.

Norman notes that this simple flow calculation can only be used for viscosity fluids of less than about 3 cP. It is a simple equation that he has found useful in oil refineries where flow is usually turbulent and pipes are made from carbon steel. However, unless you are quite sure that the flow is turbulent, it would be better to check the flow conditions in the pipe before proceeding. A pipeline that normally has turbulent flow may have laminar flow, for example, if flow rate is cut back, perhaps by closure of one or more of the plants which it is feeding. There may also be changes in temperature or even pressure which could affect the viscosity of the flowing fluid. The later sections in this chapter should help you assess the flow condition in the pipe.

It is true that in oil refineries and many chemical plants the fluids conveyed in pipes frequently fall within this low-viscosity category (less than 2 or 3 cP). Even materials we might expect to be more viscous, like the heavier hydrocarbon fractions such as atmospheric and vacuum-tower bottom products, do have quite low viscosity when pumped because it is normal practice to heat them up until the viscosity is low in order to make it easier to pump them. The pipelines are then insulated and “steam traced” to keep these heavy fluids hot and at acceptably low viscosity.

48.3 Line Sizing for Low-Viscosity and Turbulent Flow

Just as we used this simple formula:

$$\Delta P = \frac{D \times V^2}{400 \times ID}$$

earlier in this chapter to evaluate pipeline pressure drop, we can also use this same formula to check line sizes for low-viscosity fluids (less than 3 cP) in turbulent flow. The expected loss in pressure for 100 equivalent ft of horizontal pipe should be between 0.1 and 1.0 psi. If you find that the expected ΔP per 100 equivalent ft of horizontal pipe is more than 1.0 psi, then we would propose an increase in pipe diameter that would bring the ΔP per 100 equivalent ft within the range of 0.1 to 1.0 psi. The converse is also true. However, if you are designing new pipe work, you should aim for 1 psi per 100 ft.

Note that our choice of standard pipe diameter becomes more restricted for larger diameter pipe, where the increments of standard pipe diameter are greater than for small diameter pipes.

Well, I suppose that was easy enough to follow. Let's just back-track a little now so that we can put the information we have had so far in its proper context. Certainly that equation for ΔP is simple enough to use, but how do we know if it is really valid for the fluid and the flow we are dealing with? How do we know if the fluid is in turbulent flow, and for that matter, what is turbulent flow anyway? Why are most fluids piped and pumped in turbulent flow? To answer these questions, we have to back track a little, and so we should. Don't panic. Although there are many excellent and far, far more complete texts on this subject (to some of which we will refer), we will now attempt to give you a thumbnail sketch or "working guide."

48.3.1 Not All Fluids Are Created Equal

Low-viscosity materials such as we have been discussing in the previous section have flow behavior that has been classified as "Newtonian," because at constant temperature and pressure the fluid has constant viscosity, irrespective of the way in which the fluid is flowing. Other types of fluid that we will look at more closely later do not follow this straightforward flow behavior, but may exhibit varying apparent viscosity depending on the amount of shear used to move the fluid or even exhibit some kind of yield stress which must be overcome before they can begin to flow. Such fluids are known collectively as "non-Newtonian."

Let us return to the discussion on Newtonian fluids. By the way, it is worthwhile to remember that liquids and gases are both thought of as fluids.

48.3.2 Flow Regimes in Newtonian Fluids

Early pioneering work by Reynolds in 1883 showed evidence of two distinctly different types of flow patterns or "flow regimes," in circular cross-section pipes. Reynolds used a glass pipe immersed in a constant-temperature water bath fitted with a quill to introduce a single filament of colored water at the center line of the glass pipe.

At low flow rates he found that the thin, colored stream remained intact close to the center line of the pipe (see Fig. 48.3a), indicating

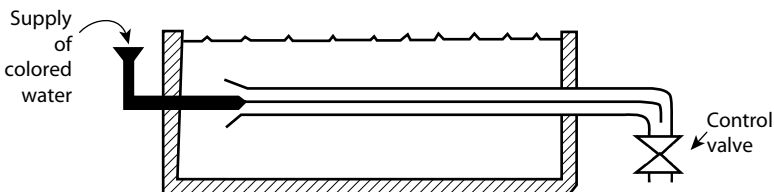


FIGURE 48.3(a) Streamline or laminar flow; the colored filament remains steady at the center line.

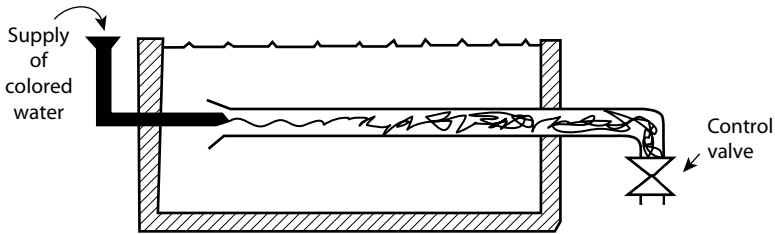


FIGURE 48.3(b) Turbulent flow develops; the colored filament oscillates and breaks up.

what we now understand to be laminar or streamline flow, which characteristically prevents much movement of the fluid in any direction except in the direction of flow parallel to the pipe. This means that there is very little mixing of the fluid as it flows, in laminar flow.

As flow rate was increased, the thin stream of colored water in the center of the pipe would begin to waver and oscillate back and forth across the pipe. The flow rate at which this occurred would vary according to the pipe diameter. Eventually, as flow continued to be increased, the colored filament would break up altogether (see Fig. 48.3b). The final stage of this flow development where the colored filament breaks up altogether and disappears we nowadays understand to be an illustration of fully developed turbulent flow. The existence of eddy currents randomly distributed across the whole cross-section of the pipe causes effective mixing throughout the fluid as it flows in all but a very narrow layer of stationary fluid, termed "the laminar sublayer," which persists next to the wall of the pipe itself (see Fig. 48.4).

The initial wavering and oscillations of the filament show that the laminar flow pattern is breaking down as eddy currents start to develop within the flowing fluid causing mixing of the fluid as it flows along the pipe. Note that the point at which the laminar flow begins to break up and become fully turbulent, as flow increases, is quite variable. We refer to this flow regime as transitional, or the

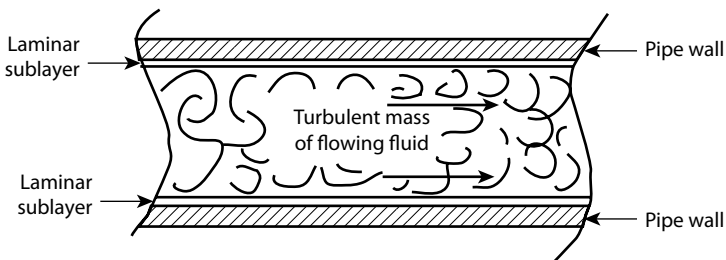


FIGURE 48.4 A narrow layer of stationary fluid, the laminar sublayer, persists next to the wall during turbulent flow.

critical zone, which gets its name from our use of the term “critical flow rate” to identify the particular flow rate that provoked turbulent flow for a given fluid in a particular pipe at certain temperature, pressure, and viscosity. Note that the onset of transitional flow depends on a number of parameters.⁴

48.3.3 The Importance of the Reynolds Number

Reynolds realized that it was not only the diameter of the pipe that affected the flow rate at which flow became transitional but also that this occurred at lower flow rates when the fluid was hot because the fluid would then have a lower viscosity. These considerations led Reynolds to develop the now-famous Reynolds number, which we may write as “Re,” and which of course has become very important in the study of fluid flow. The Reynolds number for any given flow condition has the same dimensionless value, whichever set of consistent units is employed.

$$\text{Re} = \frac{D \times V \times \text{ID}}{\mu}$$

In the foot-pound-second or fps system (which is ft–lb(mass)–s), the Reynolds number is properly calculated using the following units:

Re = Reynolds number, a dimensionless quantity

ID = Internal diameter of the pipe, ft

V = Linear velocity of the fluid at flowing temperature and pressure, ft/s

D = Density of the fluid at flowing temperature and pressure, lb/ft³
(that is, lb (mass)/ft³)

μ = Viscosity of the fluid at flowing temperature and pressure,
lb/ft × s (that is, lb (mass)/ft × s)

If you are intimidated by the prospect of the conversion of a viscosity quoted to you in the ubiquitous centipoise, or cP (which, like a cuckoo, has found its way into the American system of units but really belongs to a version of the centimeter-gram-second system), let me just show you this:

$$100 \text{ cP} = 1 \text{ poise} = 1 \text{ g}/(\text{cm} \times \text{s})$$

$$\begin{aligned} 1 \text{ cP} &= 0.01 [(1 \text{ g})^{-1} (1 \text{ cm})^{-1} (1 \text{ s})^{-1}] \\ &= 0.01 [(1/453.6) \text{ lb}]^{+1} \times [(1/30.48) \text{ ft}]^{-1} \times (1 \text{ s})^{-1}] \\ &= 0.01 [(1/453.6) \times 30.48 \times 1] \text{ lb ft}^{-1} \text{ s}^{-1} \end{aligned}$$

$$1 \text{ cP} = 6.72 \times 10^{-4} \text{ lb}/\text{ft s}$$

Incidentally, the viscosity of water at 20°C is conveniently approximately equal to 1 cP. However, the poise unit of viscosity itself was actually named after the French physician Poiseuille, who was studying blood flow and conducted extensive experiments on flow in tubes

Consistent Systems of Units					
Dimensions		SI	cgs	fps	Engineering
Length	L	meter	centimeter	foot	foot
Mass	M	kilogram	gram	pound (mass)	slug (mass)
Time	T	second	second	second	second
Force	MLT ⁻²	Newton	dyne	poundal	pound (force)

Note: 1 pound (force) = 32.2 poundal.

1 slug (mass) = 32.2 pound (mass).

1 pound (mass) under Earth's gravity produces 1 pound (weight) = 1 pound (force).

TABLE 48.1 Consistent Systems of Units

around 1840. Note that any consistent set of units will by definition provide unit acceleration to unit mass by the application of unit force. Table 48.1 provides comparative information for the SI, cgs, and engineering systems of consistent units.

The Reynolds number is a characterizer of flow type, being proportional to the ratio of the momentum of the fluid to the viscosity of the fluid. The Reynolds number also accounts for the fact that, whereas increases in momentum tend to push the fluid into turbulent flow, increases in viscosity or increases in viscous forces tend to slow up and straighten out the flow pattern.

In general, it is normal for the flow to start to change from laminar to transitional at Reynolds numbers of about 2000, and for flow to be turbulent at Reynolds numbers of above 3000. However, the precise point at which the transition will take place depends also on the roughness with respect to diameter (and geometry) of the pipe. *There is no such thing as stable transitional flow.*

It is also useful to note that a turbulent flow can become laminar if, for example, it passes into a larger diameter pipe where the Reynolds number is then less than 2000. We are also aware that in some cases unstable laminar or transitional flow has been determined for Reynolds numbers as high as 40,000, although flow at Reynolds numbers above 3000 will usually revert to being turbulent if the fluid experiences even a very small force applied at right angles to the pipe axis.

Take a look at Table 48.2 to see the effect on Reynolds numbers of changing the diameter of smooth pipes for a fixed volumetric flow rate at ambient conditions. Low-viscosity fluids are often pumped in turbulent flow simply because it would require such a large diameter pipe to maintain laminar flow that the increased cost (caused by the larger diameter) would be more than the savings generated by lower frictional losses in laminar flow.

			$Re = (D \times V \times ID)/\mu = (4/\pi) \times G/(D \times ID)$			
	Viscosity	Density	Nominal pipe diameter, inches			
	μ	D	2	6	12	14
	lb/ft s	lb/ft ³	Pipe ID, feet			
			0.17	0.5	1.0	1.1
Air	1.3×10^{-5}	0.0767	3.0×10^3	1.0×10^3	4.7×10^2	4.5×10^2
Water	6.7×10^{-4}	62.3	4.7×10^3	1.5×10^3	7.4×10^2	7.2×10^2
Gasoline	2.0×10^{-4}	46	1.1×10^5	3.9×10^4	1.8×10^4	1.7×10^4
Lube oil	1×10^{-2}	56	2.8×10^3	9.5×10^2	4.5×10^2	4.3×10^2

Notes: **Bold print** indicates laminar flow.
 $G = V \ 3 \ D \times (ID)^2 \times (\pi/4)$ pound per second.

TABLE 48.2 Effect on Reynolds Numbers of Changing Pipe Internal Diameter for a Fixed Volumetric Flow Rate at Ambient Conditions

The volumetric flow rate ft³/s used to compile Table 48.2 was selected to give a linear flow rate in the 2-inch diameter pipe of 3 ft/s. Linear flow rate (ft/s) for flow in pipes is calculated by dividing the volumetric flow rate (ft³/s) by the internal cross-sectional area (ft²) of the pipe.

48.3.4 Shear Rate, Shear Stress, and Viscosity in Newtonian Fluids

All flowing fluids experience shear, which arises from the way in which the velocity of the fluid increases as the distance from the stationary surface over which it is flowing increases. For flow in pipes, this is the distance (*y*) from the inside edge of the pipe wall. (See Figs. 48.5a and b.) The rate at which the velocity changes with respect to distance from the pipe wall is called the *shear rate* (γ), and is measured in reciprocal seconds, s⁻¹ (where [ft/s]/ft = s⁻¹).

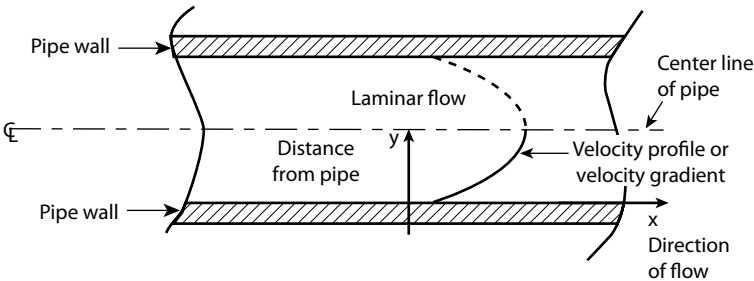


FIGURE 48.5(a) Parabolic-shaped velocity gradient laminar flow; fluid flows fastest at the center of the pipe.

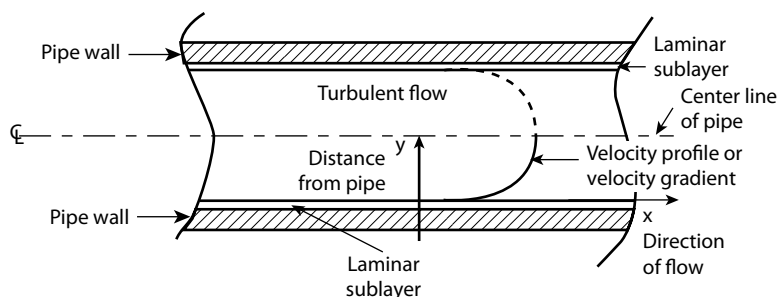


FIGURE 48.5(b) Wedge-shaped velocity gradient for turbulent flow; fluid is retarded at walls in laminar sublayer but bulk of fluid all travels at nearly the same speed.

The force per unit area of pipe surface exerted upon the liquid as a result of this velocity gradient is called the *shear stress* (R), and is measured in $\text{lb}/\text{ft s}^2$ (or poundal per square foot).

The ratio of shear stress force per unit area to shear rate is a constant for a Newtonian fluid (at constant temperature and pressure) and is called the viscosity, or sometimes also known as the shear viscosity (μ), and is measured in $\text{lb}/\text{ft s}$.

$$\text{Viscosity} = \frac{\text{Shear stress}}{\text{Shear rate}}$$

$$\mu = \frac{R}{\gamma}$$

where μ = viscosity or shear viscosity, $\text{lb}/\text{ft s}$

R = shear stress force per unit area of pipe surface, $\text{lb}/\text{ft s}^2$, or poundal per square foot

γ = shear rate, s^{-1}

Here are some typical viscosities of Newtonian fluids at ambient conditions.⁵ (The viscosity of water expressed below in $\text{lb}/\text{ft s}$ is perhaps more familiar in $\text{lb}/\text{ft h}$, that is, $6.7 \times 10^{-4} \times 3600 = 2.4 \text{ lb}/\text{ft h}$.)

	Viscosity, $\text{lb}/\text{ft s}$
Hydrogen	6.7×10^{-6}
Air	1.3×10^{-5}
Gasoline	2.0×10^{-4}
Water	6.7×10^{-4}
Lube oil	1.0×10^{-2}
Glycerol	1.0×10^{-1}
Corn syrup	1.0×10^2
Bitumen	1.0×10^8

Viscosities of liquids are generally orders of magnitude higher than for gases, and the viscosity of liquids will reduce as temperature increases, whereas the viscosity of gases will increase with increased temperature (but more slowly than liquid viscosity decreases). The higher the viscosity of the fluid at ambient conditions, the greater the rate of decrease in viscosity with temperature, for example, the viscosity of water will decrease by about 1.5 percent per 1°F at room temperature, motor oil will decrease by about 3 percent per 1°F, and bitumen will decrease by about 8 percent per 1°F.

Changes in pressure also affect viscosity, with increased pressure causing increased viscosity, although in most cases the effects are less noticeable than with changes in temperature.

For example, increases in pressure from normal atmospheric pressure to 1000 atmospheres will cause viscosity to approximately double, further increase in pressure to 2000 atmospheres might cause a tenfold increase, and increases to 4000 atmospheres could cause a 100-fold increase in viscosity. This relationship between viscosity and pressure becomes important for such applications as the lubrication of gears in a car engine where, for very short periods, the pressure between points of contact is enormous and can raise the lube oil viscosity at those points for these short times to as high as that of bitumen.⁵

48.3.5 Kinematic Viscosity

The quantity known as kinematic viscosity (kv), which we frequently see expressed in centistokes is the viscosity μ , as described above divided by the fluid density as follows.

$$\text{Kinematic viscosity} = \frac{\text{Viscosity}}{\text{Density}}$$

$$kv = \frac{\mu}{D}$$

where kv = kinematic viscosity, ft²/s

μ = viscosity or shear viscosity, lb/ft s

D = density, lb/ft³

Caution: Some texts will show you that the kinematic viscosity in centistokes is equal to the viscosity in centipoise divided by the specific gravity. This can be *numerically* correct *only* for centipoise and centistokes because the density of water (to which the specific gravity of all liquids is related) in the cgs system is 1 g/cm³, which is numerically equal to the specific gravity of water, which is also 1.0 at standard conditions. It would be far better practice to use density throughout and be certain to use consistent units in all your calculations.

We also note that because of the higher density the kinematic viscosity for a liquid is typically much lower than for a gas at atmospheric pressure. At 70°F and atmospheric pressure, the kinematic viscosity for water is about 1.1×10^{-5} ft²/s, compared to 1.7×10^{-4} ft²/s for air.⁴

48.4 Frictional Pressure Loss in Rough and Smooth Pipe

We now have to thank Stanton and Pannell, and also Moody, for their studies of flow using numerous fluids in pipes of various diameters and surface roughness and for the evolution of a very useful chart (see Fig. 48.6). This chart enables us to calculate the frictional pressure loss in a variety of circular cross-section pipes. The chart plots Reynolds numbers (Re), in terms of two more dimensionless groups: a friction factor ϕ , which represents the resistance to flow per unit area of pipe surface with respect to fluid density and velocity; and a roughness factor e/ID , which represents the length or height of surface projections relative to pipe diameter.

$$\text{The friction factor } \phi = \frac{R}{D \times V^2}$$

where R = shear stress (or flow resistance) per unit area of pipe surface, lb/ft s² (or poundal/ft²)

D = fluid density, lb/ft³

V = linear velocity, ft/s

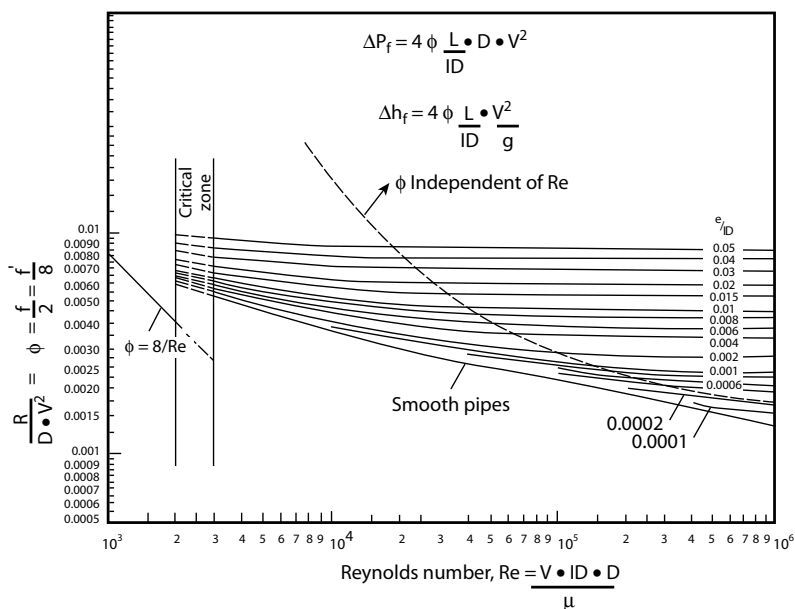


FIGURE 48.6 Friction factor ϕ vs. Reynolds number. (Reproduced by kind permission of J. F. Richardson.⁴)

Please bear in mind that two other friction factors are also in common use. These are the Fanning friction factor, f , which equals 2ϕ , and the Moody friction factor, f' , which equals 8ϕ .

$$\text{The roughness factor} = \frac{e}{\text{ID}}$$

where e = height of surface projections, ft
 ID = internal diameter of pipe, ft

Looking at the friction factor chart (Fig. 48.6), we see how the experimental data further confirms Reynolds' earlier work and helps us even today with our own field work. Once we have determined the friction factor ϕ for the pipe flow we are studying, all that remains is to evaluate the frictional pressure loss, ΔP_f from

$$\Delta P_f = 4\phi \frac{l}{\text{ID}} \times D \times V^2$$

where l = length of the pipe, ft
 ID = pipe internal diameter, ft
 D = fluid density, lb/ft³ (or lb (mass)/ft³)
 V = linear velocity, ft/s
 ΔP_f = frictional pressure loss, lb/ft⁻¹ s⁻², or poundal/ft²
 (1 poundal = 1 lb_(mass) × ft × s⁻²)

Naturally, you will need to know the relationship between the poundal/ft² [or lb (mass) × ft⁻¹ s⁻²] and the familiar psi, in order to put this to everyday use. Here it is:

$$\begin{aligned} 1 \text{ poundal/ft}^2 &= (1/32.2) \text{ lb}_{(\text{force})} \times \text{ft}^{-2} \\ &= (1/32.2) \times (1/144) \text{ psi} \end{aligned}$$

Or, if you prefer, the frictional head loss in feet of liquid:

$$h_f = 4\phi \frac{l}{\text{ID}} \times \frac{V^2}{g}$$

where h_f = frictional head loss, ft
 g = acceleration due to gravity 32.2 ft/s²

All other symbols and units are the same as used in preceding sections.

We must also note that it is not really possible to make reliable evaluations of expected frictional pressure losses within the transitional or critical flow regime that exists between Reynolds numbers of 2000 and 3000.

48.4.1 Values of Roughness

The roughness e of a pipe does not depend on the nature of the fluid being conveyed through the pipe. Once the e value for a given length of pipe is known, then the same value is used whatever liquid or gas is in the pipeline, as long as there is no further corrosion or damage to the inside of the pipeline.

Most often we have to assume a value for pipe roughness e by using standard values from tables. Here are some examples:

	Roughness e , ft
Drawn tubing	0.000005
Commercial steel and wrought iron	0.00015
Asphalted cast iron	0.0004
Galvanized iron	0.0005
Cast iron	0.00085
Wood stave	0.0006–0.003
Concrete	0.001–0.01
Riveted steel	0.003–0.03

But if you are able to pump water through the pipeline, at a velocity sufficiently high to give rise to turbulent flow ($Re > ca\ 2500$, but the higher the better), then the pipe roughness e can be found by calculating the Reynolds number and measuring the frictional pressure drop or frictional head loss along the pipe. Then using the friction factor ϕ and Reynolds number Re you can find the roughness e via the relative roughness term e/ID from the friction factor chart as illustrated in Fig. 48.6.

For example: There is a 10-inch steel pipeline that is used to deliver various liquid products from a production facility to a dock. The line is flushed with water when switching products. There have been questions as to whether some older sections of this line need to be replaced because of increased internal roughness. The first part of the line is the oldest part and runs up over a hill and in an area overgrown with bushes and thorns, quite difficult to inspect. We check the installation drawings to find the length of the line between the two closest pressure points, and find the elevations of the pressure points from a large-scale contour map of the area. We measure the flow rate with water in the line by monitoring the change in the level of water in the supply tank; this may require a very large tank for high flow rates.

The calculated head loss is given by:

$$h_f = 4 \phi \frac{l}{ID} \times \frac{V^2}{g}$$

where l above is lowercase L (sometimes this can be confusing in printed text) and refers to the length of the section of line (ft) of 8340 ft.

ID = 10 inch \equiv 0.833 ft

P1 = inlet pressure for this section of line = 35 psig

P2 = outlet pressure for this section of line = 23 psig

Elevation of P1 is 250 ft above sea level.

Elevation of P2 is 160 ft above sea level.

Elevation of the highest point in the line is 310 ft above sea level.

$V = 7$ ft/s

h_f is the head loss in feet of liquid, in this case feet of water = 118 ft

$$\text{Reynolds No.} = \text{Re} = \frac{D \times V \times \text{ID}}{\mu} \equiv \frac{62.3 \times 7 \times 0.833}{0.00067} = 5.4 \times 10^5$$

where D for water = 62.3 lb/ft³

μ for water (at 70°F, 1 atm) = 6.7×10^{-4} lb/ft.s

And the friction factor ϕ is found from:

$$\phi = \frac{R}{D \times V^2} = \frac{h_f}{4} \times \frac{\text{ID}}{1} \times \frac{g}{V^2}$$

$$\phi = \frac{118}{4} \times \frac{0.833}{8340} \times \frac{32.2}{7 \times 7} = 0.0019$$

where $g = 32.2$ ft/s²

Referring to Fig. 48.6, we find $e/\text{ID} = 0.0003$.

The roughness, $e = 0.0003 \times 0.833 \equiv 0.00025$ ft, or if we look on a more detailed chart,⁴ this value for e/ID may be found to be close to 0.0004 to give e as 0.00033, which still compares with values quoted for steel pipe, but the internal surface is becoming rough, as a result of corrosion erosion or adhering deposits. However, please note that this approach is an approximation. Generally the effects of aging are not likely to be uniform along the whole pipeline, so that e/ID may vary along the length of the pipe. Particularly, one might expect a degree of pitting and build up of local corrosion and other debris inside the pipeline at a number of critical points such as bend, fittings, and so forth.

48.5 Special Case for Laminar Flow

You will have noticed that there is just one line on the friction factor chart (Fig. 48.6) for ϕ in terms of Re in the laminar flow region (Re less than 2×10^3) of slope $\phi = 8/\text{Re}$. This means that for Reynolds numbers of less than 2000 we can very simply calculate the frictional pressure loss by the relationship

$$\Delta P_f = 32 \frac{\mu \times V \times l}{\text{ID}^2}$$

where l = length of pipe, ft

ID = internal diameter of pipe, ft

D = density of fluid, lb/ft³

V = linear velocity, ft/s

ΔP_f = frictional pressure loss, poundal/ft² (or lb (mass) ft⁻¹ s⁻²)

[1 poundal = 1/(32.2 × 144) psi]

48.6 Smooth Pipes and Turbulent Flow

Another simple relationship was developed by Blasius for flow in smooth pipes at Reynolds numbers between 2.5×10^3 and 1×10^5 . In this region the friction factor ϕ for smooth circular cross-section pipes approximates to

$$\phi = 0.04 \text{ Re}^{-1/4}$$

which gives us

$$\Delta P_f = 0.16 \times \frac{l \times D \times V^2}{\text{ID} \times \text{Re}^{-1/4}}$$

All the symbol units used here are the same as in preceding sections, and ΔP_f is once again expressed as poundal/ft², or lb (mass) ft⁻¹ s⁻².

48.7 Very Rough Pipes and Very Turbulent Flow

The friction factor ϕ becomes independent of the Reynolds number and tends toward a constant value depending only on the value of the relative roughness (e/ID) see Fig. 48.6.

Suppose e/ID is 0.05, then for values of Re above 1×10^5 the friction factor ϕ equals 0.0087 (see Fig. 40.6), hence in this case

$$\Delta P_f = 4 \times 0.0087 \frac{l}{\text{ID}} \times D \times V^2$$

$$\Delta P_f = 4 \times 0.035 \frac{l}{\text{ID}} \times D \times V^2$$

All the symbols and units used are the same as in preceding sections.

48.8 Non-Newtonian Fluids

We have defined a Newtonian fluid as one that has a constant viscosity at a given temperature and pressure irrespective of the way in which it is flowing or being moved. We can also say that the ratio of shear stress to shear rate in such a fluid would have a constant value at constant temperature and pressure. Effectively, all gases approximate to this Newtonian type of flow behavior and most low-viscosity liquids are also Newtonian. That a fluid is either compressible (like a gas)

or incompressible (like water) has no significance in deciding whether it is Newtonian or non-Newtonian in flow behavior.

Non-Newtonian fluids show a different apparent viscosity depending on the shear to which they are exposed—that is, depending on the way in which they are flowing or being moved. Apparent viscosity for some fluids varies with the magnitude of the shear, and for others it depends also on how long they experience shear. Usually non-Newtonian flow arises in fluids with complex molecular structures (e.g., polymers) and some hydrocarbon crude oils, and where large groups of particles are loosely held together such as in flocculated slurries or suspensions of solids in liquids; pastes and emulsions also usually fall into this category.

To decide whether a fluid is Newtonian or non-Newtonian, we need to know how it behaves in response to shear.

Suppose I now explain that the Newtonian kind of flow behavior is really just a special case, just a way that some fluids behave over a particular range of shear rates. You can see it has been suggested that at high enough shear rate, *all* liquids become non-Newtonian.⁵ It is just that the liquids we have been considering earlier in this chapter and elsewhere in this book only become non-Newtonian at very high rates of shear. For example, the critical shear rate for glycerol and mineral oils above which they cease to behave as Newtonian liquids would be above about $1 \times 10^5 \text{ s}^{-1}$. It has also been estimated that pentane becomes non-Newtonian above shear rates of $5 \times 10^6 \text{ s}^{-1}$, and it has even been suggested that water would start to show non-Newtonian behavior⁵ at the impossibly high shear rate of 10^{12} s^{-1} . So what kind of flow behavior can we expect from the fluids around us? Some fluids may even appear at first glance to be more like solids than liquids and not seem really capable of “flow” in the ordinary sense at all. Fortunately a number of very dedicated people have also been asking this same question for many years, and there is now a wealth of experimental information, mathematical modeling, and excellent texts to help us come to terms with this situation.⁴⁻⁶ But while we are discussing some of the simplest forms of non-Newtonian behavior, please be aware that most fluids will exhibit more than one type of non-Newtonian behavior. In these circumstances we usually select the most dominant characteristics to predict the flow behavior.

48.9 Some Types of Flow Behavior

Take a look at Fig. 48.7. This should give you an idea of some of the variations of fluid flow behavior. The gradients or slopes of these lines or curves are variable within the identified flow regime. It is whether or not the curves pass through “the origin” and the general shape of the curve that defines the type of behavior.

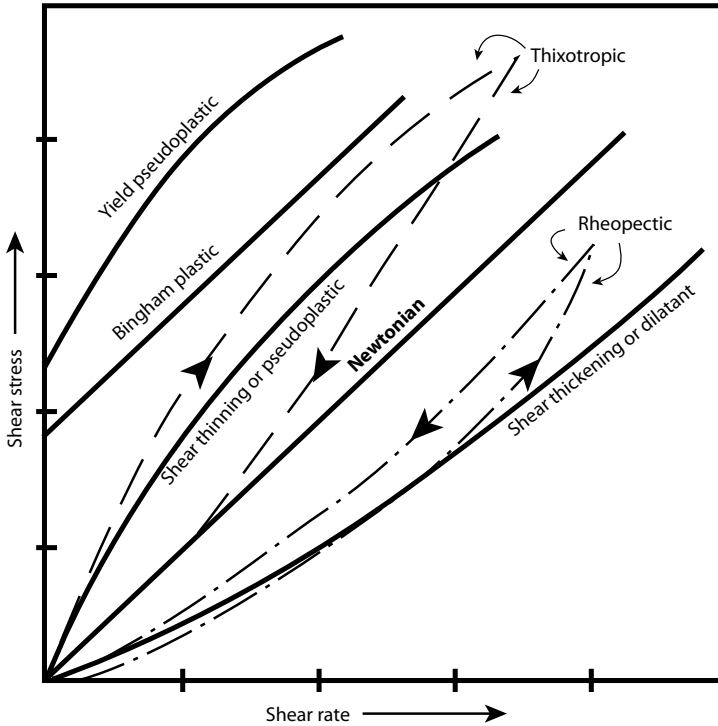


FIGURE 48.7 Examples of variations of fluid flow behavior.

The diagram shown in Fig. 48.8 is also included to guide you as we take a little closer, qualitative look at some of these types of non-Newtonian flows (for more detailed information we suggest you begin with the references 4, 5, and 6 at the end of this chapter).

48.9.1 Time-Independent Flow

As the name suggests, this type of flow behavior is not affected by the length of time that the fluid has been flowing or being moved. We say it is unaffected by the duration of the shear which it experiences. This type of flow is the best documented, and some simple models of predicting flow exist to which we will later refer.

In this group we have Newtonian flow with which we are familiar, shear-thinning, shear-thickening, and viscoplastic flow. Please note once again that these non-Newtonian types of flow may be exhibited by fluids that will indeed also behave as Newtonian liquids over a particular range of shear rates.

Shear-thinning liquids in general become more fluid, less viscous, as the shear rate on them increases. Typical examples of such fluids are: shower gel, hair conditioner, fabric conditioner, liquid abrasive cleaner, fabric washing liquid, printing ink, facial wash, flocculated

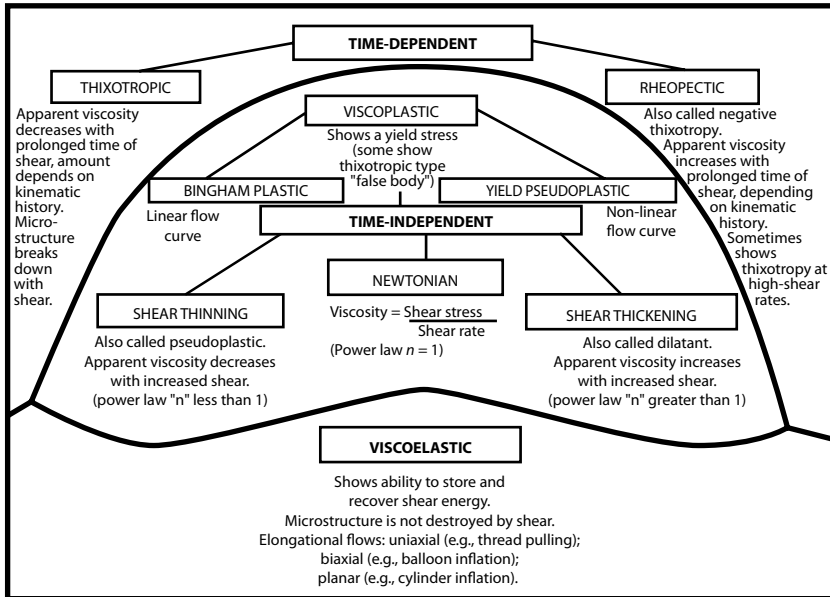


FIGURE 48.8 Some types of fluid flow behavior.

clay slurries, 1 percent aqueous solutions of food thickeners (such as guar gum, locust bean gum, carrageenan, xanthan gum, alginate), most paints, many 1 to 2 percent aqueous polymer solutions, and some hydrocarbon crude oils.

In particular, with fluids such as flocculated clay slurries, which may have very high apparent viscosity at low shear rates, the fact that they are shear thinning enables us to pump them and convey them relatively easily by pipeline because of the high rates of shear applied by the fast spinning impeller of a centrifugal pump and the shear that they experience at the pipe wall (and another phenomena known as wall slip that also reduces the viscosity at the pipe wall).

Shear-thickening liquids are liquids whose apparent viscosity increases with increased shear rate. The old name for this is "dilatant" because the fluid appears to swell as it is sheared. The best way I can describe this is to ask you to think of a bucket of sand and water with about an inch of water above the settled sand. (Think back to your childhood visits to the seaside or sand pit.) Suppose you now rock the bucket from side to side; the sand and water mix together to form a rather unstable suspension. However, if you take a stick and try to stir the contents of the bucket with any appreciable amount of force it is virtually impossible. The suspension actually begins to look like a fractured solid. This is an extreme example of shear thickening. The voidage between the sand particles and the amount of water present when the mixture behaves like a suspension is just enough to

allow the mixture to flow at very low shear rates and let the particles move past each other. At high shear rates, the mixture expands, there is not enough water to fill the voids, so the particles of sand rub against each other, dramatically increasing the apparent viscosity as the shear rate increases. Other materials that exhibit this type of behavior are concentrated (especially deflocculated) aqueous suspensions of china clay, titanium oxide, corn flour, and wet cement aggregates.

Viscoplastic fluids are those that appear to show a “yield stress.” That is, a certain amount of shear stress must be applied to the material before it can begin to flow, or deform. Whereas in fact it seems that if flow is measured over a wide enough range of shear rates, no yield stress really does exist,⁵ the concept of a yield stress remains very convenient, because a number of materials do closely approximate to this type of behavior.

If the material is being sheared but at less than the yield shear stress (R_{yield}), then the material will deform “elastically” and flow as a rigid body and not like a fluid. The classic example of this type of behavior is the behavior of toothpaste, which flows “en masse” through the nozzle of the container tube when the tube is squeezed (shear at the nozzle wall being much higher than in the bulk of the paste), but stays firmly rod shaped on the toothbrush until at some higher shear rate it deforms when brushed onto one’s teeth. Actually, this is an example of a *Bingham plastic*, which, once the shear stress exceeds the yield stress for a limited range of shear rates, will then flow with shear stress directly proportioned to shear rate. To be more precise, we should really say that such fluids flow with any further shear stress (R), over and above the yield stress, in a way that is directly proportioned to shear rate. This is somewhat similar to Newtonian behavior except that the yield shear stress (R_{yield}) must be reached before there can be any flow. After that yield stress has been reached and over a limited range of shear rate ($\dot{\gamma}$), we have:

$$[(R - R_{\text{yield}}) / \dot{\gamma}] = \text{constant}$$

We also find fluids that appear to have a yield stress but once that is reached will then show shear-thinning flow behavior, and these we call *yield pseudoplastic*. These fluids are also sometimes known as Herschel-Bulkley fluids and for these we can write:

$$[(R - R_{\text{yield}}) / \dot{\gamma}^n] = \text{constant}$$

This covers a wider range of fluids than that of the Bingham plastic and so there are numerous examples of these yield pseudoplastic materials quoted in texts, including: mayonnaise, ketchup, some printing inks, molten chocolate, plastic explosives, blood, water-coal mixtures,⁵ foams, battery and rocket propellant pastes, and drilling muds.

48.9.2 Time-Dependent Flow

These are fluids whose apparent viscosity changes according to the *length of time that they are exposed to shear*, as well as in accordance with the rate of shear. We also note that if a fluid of this type has already been exposed to shear, by pumping or mixing, and so on, it is likely to have a different apparent viscosity, or rather, “effective viscosity,” than that of the virgin unsheared material. Finally, just to complete the picture, if a material of this type is exposed to shear (e.g., pumping or mixing) and then allowed to stand or “relax” in a tank, for example, then to some extent and once again depending on how long the material has rested since being sheared, its effective viscosity will be restored to some degree, and in some cases, close to that of the virgin material. The past treatment of the material with reference to the degree of shear experienced and time of relaxation, and so forth, is known as the *kinematic history* of the fluid.

Thixotropic fluids are time-dependent shear thinning in nature although for any given shear rate as shearing continues, an equilibrium will eventually be established and the viscosity will reach a constant value. The explanation for this is that the microstructure of the material breaks down with shear, and to some extent, is able to rebuild when the material is left to stand. The best example of a thixotropic material is nondrip (thixotropic) house paint. It looks almost gelatinous or even semisolid in the paint can, but flows like a liquid when stirred rapidly. It stays on the brush between paint can and house wall (while there is no shear), and then flows over the wall surface when sheared by the action of the brush strokes.

Rheopectic fluids are sometimes said to behave with negative thixotropy, meaning that they are time dependent as is the case with thixotropic fluids, but are instead shear thickening in nature. This type of behavior has been documented⁵ for a 42 percent aqueous gypsum paste, which when shaken and then rested for 40 min resolidified. But this occurred in just 20 s if the paste was only gently sheared (i.e., the container was rolled gently between the palms of the hands).

Once again, please bear in mind that it is not unusual for a material to exhibit both thixotropy and rheopexy depending on the shear rate and concentration of solids.

48.10 Viscoelastic Fluids

These are fluids that have the ability to store and recover shear energy and, hence, show some properties of a solid and some properties of a liquid. There are two easily observable examples of viscoelastic behavior:

- *The soup bowl effect:* Suppose we gently stir a bowl of soup with a spoon so that the soup revolves around the bowl. If the spoon is lifted out of the soup (i.e., the energy source is removed), the

soup will gradually come to rest (the inertial circulation dies out due to the effects of viscous forces), and then if the soup is viscoelastic (as many canned soups are), it may seem to rotate in the opposite direction or “unwind” a little.

- *Die swell*: The jet formed by a viscoelastic fluid emerging from a nozzle or die may be of larger diameter than the aperture through which it flows. This is due to the removal of a constraining force on the fluid.

Other examples of viscoelastic materials are synovial fluid, molten polymers (with thread-forming properties used in fiber spinning or film blowing), and “bouncing putty” or “nutty putty,” which will flow if stretched slowly, but bounces if struck hard against a hard surface.

48.11 Identifying the Type of Flow Behavior

Depending on the kind of process you work with, you may have facilities on hand to check the rheology (flowing characteristics) of the fluids involved in your work. If so, you can plot the shear stress against shear rate over your working range of shear rates to see which type of flow behavior you have as compared to those shown in Fig. 48.7. There are many different kinds of instruments (viscometers) to measure these properties. Further discussion of these may be found in references 4, 5, and 6 at the end of this chapter.

Otherwise, bear in mind that very often the range of shear rates to which a process fluid is exposed in pipe flow is often quite limited, and because of this it is often possible to adequately represent the flow behavior of a process fluid over a limited range of shear rates by the power law or Ostwald de Waele model, and the Metzner-Reed Reynolds number,⁷ which we shall shortly discuss. Meanwhile, in the absence of any lab equipment to provide you with shear-stress versus shear-rate data, we suggest that you vary the flow rate to provide several flow rates, and run pressure drop surveys over the pipeline in question. Tabulate the data and use this to develop a “power law” relationship (see following sections) as a first approximation for your pipeline and process fluid. Most likely you will not need to perform any more elaborate study than that.

48.12 Apparent and Effective Viscosity of Non-Newtonian Liquids

We have said that for a Newtonian fluid the viscosity can be expressed as shear stress divided by shear rate:

$$\mu = \frac{R}{\gamma}$$

which is a constant for constant temperature and pressure.

When we consider non-Newtonian liquids, the viscosity is not constant, because of the influence of shear rate and time. So for non-Newtonian fluids we refer to their apparent viscosity (μ_a), or effective viscosity, and we have to specify the conditions of shear rate to which this applies. When we speak of time-dependent fluids, we must also specify the shear history or kinematic history, at the point that their apparent viscosity is quoted. As long as we keep these restrictions in mind for a non-Newtonian fluid, we can still think of the apparent viscosity as:

$$\mu_a = \frac{R}{\dot{\gamma}}$$

That is, the apparent viscosity of a non-Newtonian liquid can be expressed as the ratio of shear stress to shear rate at a specific point within the fluid, and perhaps even at a specific point in time, or within a specific shear-rate range, depending on the nature of the liquid.

Effective viscosity is a term used to relate non-Newtonian behavior to that of some hypothetical equivalent Newtonian fluid, just as is apparent viscosity. However, the term *effective viscosity* really relates more to the behavior of the bulk of the fluid and is, as a concept, equal to the Newtonian viscosity that would give the same relationship.⁶

The units of measurements of apparent viscosity or effective viscosity are the same as those used for viscosity when we refer to a Newtonian fluid. We have mainly used lb/ft s, which is the unit of viscosity in the fps system.

48.13 The Power Law or Ostwald de Waele Model

This was conceived as a means of predicting the flow behavior of time-independent fluids, particularly shear-thinning and shear-thickening fluids. It is really nothing more than an equation to fit the plot of shear stress versus shear rate for such a fluid when both are plotted on logarithmic scaled paper.

The basic form of this is

$$R = k (\dot{\gamma})^n$$

from which the apparent viscosity, at any given point within the fluid

$$\mu_a = \frac{R}{\dot{\gamma}} \text{ becomes } \mu_a = k(\dot{\gamma})^{n-1}$$

where k is known as the consistency coefficient

n is the power law index

Both k and n must be experimentally derived for each fluid that one is working with over the appropriate (operable) range of shear rates. In pipe flow this translates to running a series of different flow rates through the same pipe system as will carry the fluid and measuring

pressure drop for each flow rate in order to derive k and n , as we shall see later. It is also possible to measure k and n by plotting shear stress versus shear rate using data from laboratory viscometer measurements on the fluid with which you are working.

In general if:

- n is less than 1, the fluid shows shear thinning.
- n equals 1, the fluid behaves as Newtonian.
- n is more than 1, the fluid shows shear thickening.

The smaller the value of n , the more shear thinning the liquid will be. The opposite is true for a shear-thickening liquid.

The whole idea is that for these types of fluids some portion of their shear-stress versus shear-rate plot on log scales would tend to be a straight line. (Just as Newtonian behavior where $n = 1$ would provide a straight line for such a plot whether plotted on log scales or ordinary graph paper.) The fact that such plots on log scales (where n is not equal to 1) are straight lines only over some portion of the shear-rate range is the reason why it is necessary to stringently limit the range of shear rates over which you apply the equation to match those for which the values of n and k were determined. *Do not extrapolate beyond known experimental data for the liquid with which you are working.*

Please note that values of k cannot be compared for fluids with different n values. This is because the value and units of the consistency index k depend upon the value of the power law index n . (In fact k actually has dimensions $\text{ML}^{-1} \text{T}^{n-2}$, whereas n is dimensionless.)

The areas where this model (equation) will be the most unreliable are at very low and very high rates of shear. It has been suggested⁵ that this model is typically at its most reliable over shear-rate ranges between 10^1 and 10^3 s^{-1} .

The values of k and n may not always be the same across the whole range, depending on the fluid. However, it is possible to deal with this and sometimes even larger ranges of shear rates by breaking the range into smaller sections. Then deal with the whole range section by section using appropriate values of k and n .

In general, even with its limitations, the power law model is usually the first choice for engineering calculations because it is simple, and when used with care, over a narrow or medium range of shear rates, it provides adequate and quite reasonable predictions.

48.13.1 Evaluating k and n

The classical way to evaluate k and n for a power law fluid is by using \log_{10} scales to plot the shear stress (in fps units, that is, poundal/ft² or lb (mass)/ft s² on the vertical axis, versus nominal shear rate (s^{-1}) on the

horizontal axis. (See Fig. 48.9.) Using the same symbols and units as in earlier sections, this is:

$$\text{Nominal shear rate} = \frac{8V}{ID}$$

$$\text{Shear stress at wall} = \frac{\Delta P}{L} \times \frac{ID}{4}$$

If all the points you plot fall close to a single straight line, you are assured that the fluid fits the power law over the range of shear rates that you are using.

The power law index n can then be found from the slope of the line (see Fig. 48.9) and k is found for fluids in laminar flow via k as follows,⁷ using the value obtained for n .

$$k = [(ID \times \Delta P)/4I] / [(8V)/ID]^n$$

where $n = n$ for power law liquids
and then:

$$k = \frac{k}{\frac{(3n+1)^n}{4n}}$$

There are many different kinds of viscometers in use for measuring these properties. Perhaps the best for power law fluids would be

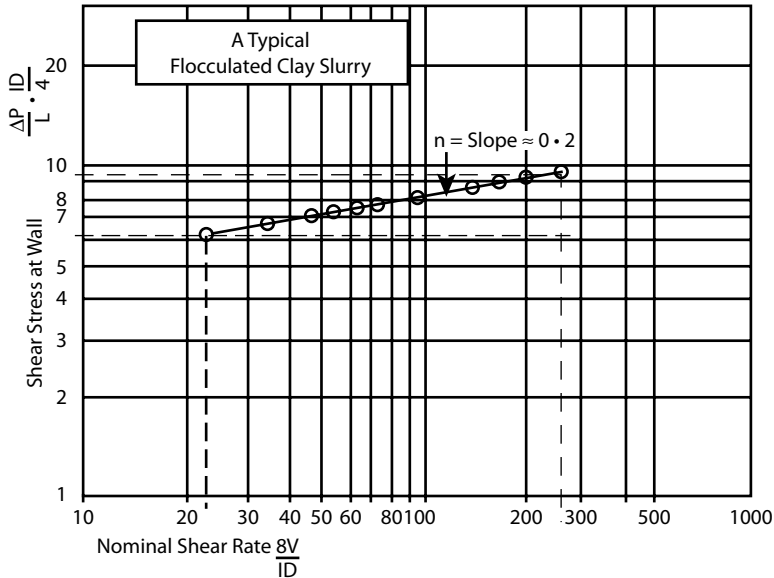


FIGURE 48.9 Evaluating the power law index n from experimental data.

either the concentric cylinder viscometer or the tube viscometer; please refer to references 4, 5, and 6 for details of these.

Alternatively, over *very* small ranges of flow rate you could try using two pairs of flow-rate and pressure-drop measurements (one pair at each end of your flow-rate range), and use the following relationship with both pairs of measurements in succession, to find k and n directly, for fluids in laminar flow.

$$V = [(\Delta P)/(4kl)]^{1/n} \times [n/(6n+2)] \times [ID]^{(n+1)/n}$$

48.14 Generalized Reynolds Numbers

In order to be able to use the same friction factor data for non-Newtonian fluids as has been developed for Newtonian fluids, generalized Reynolds numbers of the following form have been developed.

$$\text{Re}_{\text{gen.}} = \frac{D \times V \times ID}{\mu_{\text{effective}}}$$

48.14.1 The Metzner-Reed Reynolds Number, Re_{MR}

The Metzner-Reed Reynolds number⁷ is the generalized Reynolds number to use when working with the power law. In order to make comparison with other texts and charts easier for you, I will write it out in its usual form; please bear with me, it really is not complicated.

$$\text{Re}_{\text{MR}} = \frac{D \times V^{2-n} \times ID^n}{8^{n-1} \times k}$$

For the power law application⁷ of Re_{MR}

$$n = n \text{ and } k = k [(3n+1)/4n]^n$$

By the way, this also provides us with another expression for apparent viscosity or effective viscosity, as follows:

$$\mu_{\text{effective}} = k [(8V)/ID]^{n-1}$$

48.14.2 Friction Factor and Re_{MR}

Just as we had a friction factor ϕ for Newtonian fluids, we have the same commodity for non-Newtonian power law fluids:

$$\phi = \frac{R}{D \times V^2}$$

We also have the Fanning friction factor, f , which equals 2ϕ ; and the Moody friction factor f' , which equals 8ϕ , just as we saw earlier when discussing frictional pressure loss in rough and smooth pipe for Newtonian fluids.

The Metzner-Reed Reynolds number can also be used in conjunction with the familiar expressions for frictional pressure loss (ΔP_f) and head loss (h_f), just as with Newtonian flow as seen earlier in this chapter:

$$\Delta P_f = 4\phi \frac{l}{ID} \times D \times V^2$$

and

$$h_f = 4\phi \frac{l}{ID} \times \frac{V^2}{g}$$

In *laminar flow* for power law fluids in smooth or rough pipes, we can still use⁴ the familiar:

$$\phi = 8/\text{Re}_{\text{MR}}$$

The standard friction factor chart (Fig. 48.6) can also be used with Re_{MR} . Please note that two liquids having similar k and n values in laminar flow may not necessarily behave similarly to one another in turbulent flow, as would be the case if you compared a flocculated clay suspension with a polymer solution.

In *turbulent flow* for smooth pipes we can use an expression derived from the Blasius equation:

$$\phi = \frac{R}{DV^2} = 0.0396 n^{0.075} \times \text{Re}_{\text{MR}}^{-0.25}$$

where n is the power law index.

This expression gives the friction factor to within plus or minus 10 percent accuracy. Other relationships do exist,⁶ but they are more complex and do not provide very much greater accuracy. However, even to predict ΔP under turbulent conditions one needs to have evaluated k and n from laminar flow data at the same wall shear stress as that expected in the actual pipe flow operating under turbulent conditions.⁶

In general, non-Newtonian flow behavior (or rheology) is of dominant importance in laminar flow because it influences viscous shear. The transition from laminar to turbulent flow is much less sharply defined for non-Newtonian fluids than it is for Newtonian, and it is difficult to predict with any degree of accuracy. However, a value of the Metzner-Reed Reynolds number (Re_{MR}) of 2000 is still a reasonable estimate for the transition from laminar to turbulent flow.

When turbulence takes over, the non-Newtonian rheology plays a much less important role in determining how the liquid will flow. In fact, there is not likely to be too much error by treating shear-thinning fluids

in turbulent flow as if they were Newtonian. Therefore, the turbulent flow of a shear-thinning fluid can be reasonably approximated using Re_{MR} and the friction factor chart shown in Fig. 48.6.

There is also a friction factor chart drawn by Dodge and Metzner⁸ plotting the Fanning friction factor versus the Metzner-Reed Reynolds number (although this has been subsequently redrawn for convenience with ϕ , see ref. 4). This chart shows that the friction factor decreases with n in turbulent flow—that is, it decreases as shear-thinning character increases. If you are studying this chart, be aware that the only experimentally verified data for turbulent flow is that given for $n = 1.0$ to $n = 0.4$.

The material presented in this chapter has been used by the author to study and predict the flow behavior of water and solid-liquid suspensions in a 10-mi-long pipeline between a processing plant and dock facility. I would tell you about it now, but that's another story, and this chapter is long enough already!

If you are trying to predict non-Newtonian flow behavior, this chapter should give you a place to start. The references may also be useful and they will in turn provide you with many more references, historic and recent. This is a developing field.^{5,6} At present, there is increasing demand to predict how slurries, emulsions, gels, pastes polymers, heavy oils, and so on, will flow and how they will behave if left to stand over long periods.

Therefore, if you are trying to do more precise work with non-Newtonian fluids I would advise you to try as many of the correlations as you can find in the literature⁷⁻⁹ that are relevant to the fluid and conditions that you are working with, and then compare the results!

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CHAPTER 49

Super-Fractionation Separation Stage

The concept of the super-fractionation separation stage in distillation is a key concept in process operations and chemical engineering. Yet I have never seen this idea discussed in any text concerning distillation.

The effect is observed in two areas of a distillation operation:

- Partial condensation
- Once-through thermosyphon and kettle reboilers

The concept of a super-fractionation tray is important when separating products, if either the top or bottom product consists of a multi-component mixture. For example, if we are fractionating only a mixture of propane and isobutane, the super-fractionation concept would not be significant. However, if the overhead product consisted of a mixture of ethane, propane, and isobutane (i.e., C_2s , C_3s , and C_4s) and the overhead condenser was a partial condenser (see Fig. 49.1), then the partial condenser would serve as a super-fractionation stage. The partial condenser would contribute far more to fractionation than any of the other stages in the upper portion of the distillation tower.

If the bottom product consisted of a mixture of isopentane, normal hexane, and normal heptane (i.e., C_5s , C_6s , and C_7s) and the reboiler was a kettle-type reboiler (see Fig. 49.2), then the kettle reboiler would serve as a super-fractionation stage. The kettle reboiler would contribute far more to fractionation than any of the other stages in the lower portion of the distillation tower.

49.1 My First Encounter with Super-Fractionation

In 1969, I was working for Amoco Oil as a process design engineer. One of my earliest designs for Amoco was a gasoline debutanizer, the bottom portion of which is shown in Fig. 49.3. Note that this tower is

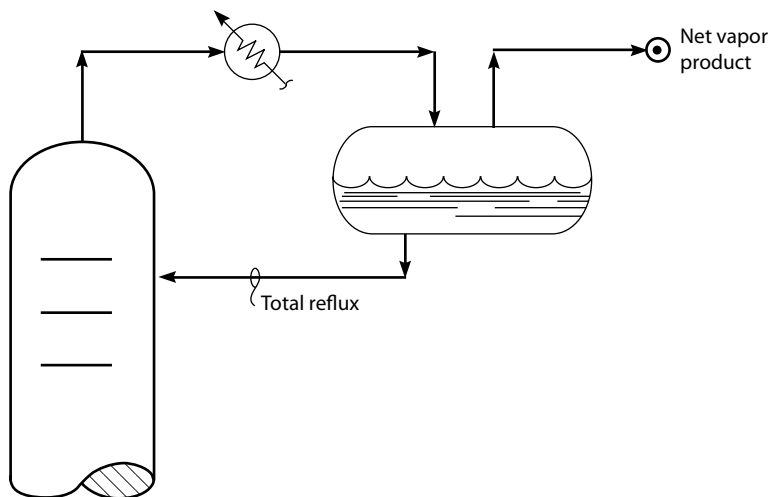


FIGURE 49.1 A partial condenser serves as a super-fractionation stage.

equipped with a once-through thermosyphon reboiler. Such a reboiler works as the bottom theoretical stage of the debutanizer assuming that:

1. All of the liquid from the bottom tray flows to the reboiler.
2. Valve A is closed.

But valve A was not closed. I had opened the valve myself during the initial tower start-up. Opening valve A is a necessary part of the tower start-up procedure. This allows liquid to gravitate out of the bottom of the tower and into the reboiler, the objective being to establish

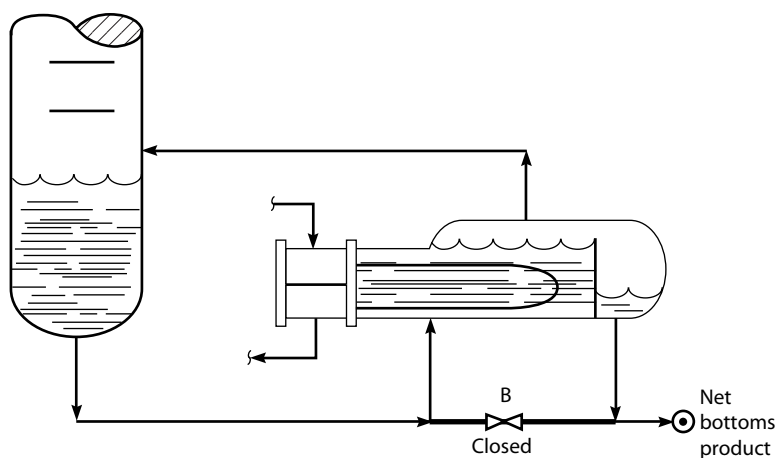
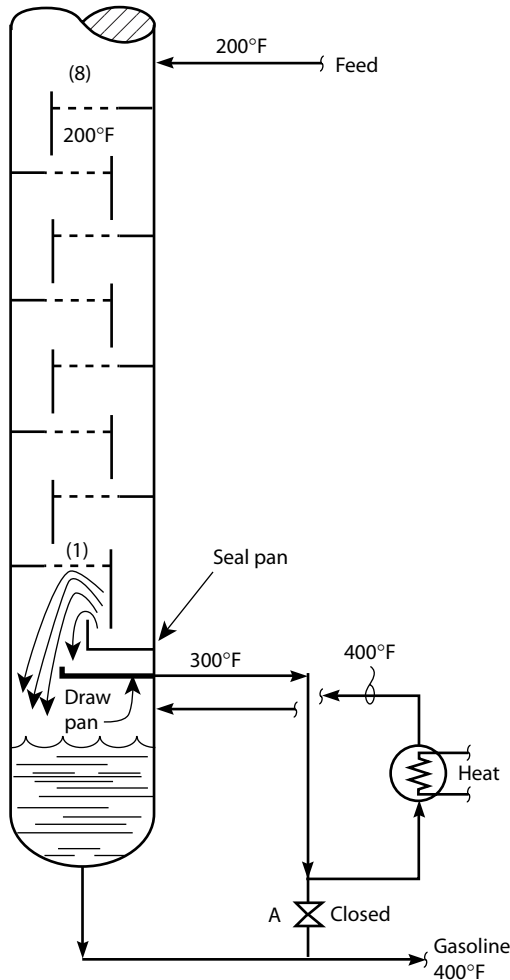


FIGURE 49.2 A kettle reboiler serves as a super-fractionation stage.

FIGURE 49.3

Once-through thermosyphon reboiler is also a super-fractionation stage.



thermosyphon circulation until a vapor flow is established through tray deck 1. However, when I tried to close valve A, the reboiler duty died off, indicating a loss of flow into the shell side of the reboiler. So, I was forced to leave the start-up line (i.e., valve A shown in Fig. 49.3) open to sustain the reboiler duty.

The reboiler now seemed to work fine. The problem was that the debutanizer tower was not doing a good job of debutanizing. The butane content of the gasoline bottoms product was 4 percent. My design for the tower was based on 1 percent butane in gasoline. Something was terribly wrong.

I rechecked my computer model for the tower. With an assumed 80 percent tray efficiency, which is normal for this service at the observed reflux ratio, a 1 percent butane in the gasoline would be expected. To force my computer model to match the observed 4 percent

butane in the gasoline, I had to assume a low 40 percent tray efficiency for the bottom eight trays. I concluded something was wrong with these trays.

My boss organized a meeting with the plant manager. I reported that the bottom eight trays must have been damaged on start-up as indicated by their low tray efficiency of 40 percent, as compared to their design tray efficiency of 80 percent.

The plant manager was doubtful. Perhaps, he asked, this could be an installation error? Perhaps, he wondered, the trays had not been installed properly?

"No, that's not possible," I replied, as I had personally inspected the installed trays and approved the tower as ready for closure. No, the trays were designed and installed properly. They must have been damaged on start-up. I advised that we should now order new replacement trays. We should shut down the debutanizer as soon as the replacement trays arrived.

When the new trays were delivered, the debutanizer was taken offline, steamed out, and opened. When I entered the tower, I discovered that each and every tray in the column was totally and completely intact and clean. Everything was just as I had last seen it six months ago. Actually, the only thing that had changed was me. I had become smarter in the last six months. For example, I now noticed that the sides of the draw pan (see Fig. 49.3) were missing. True, they had been missing last year too, but I had not noticed their absence then.

This explained why I had been forced to open valve A (the start-up line) to get flow to the reboiler. That is, the liquid from the bottom tray seal pan was largely overflowing the draw pan and bypassing the reboiler. Hence, the only way I could get enough liquid into the reboiler was to gravitate it out of the bottom of the tower through valve A.

But could this explain the poor fractionation efficiency in the bottom of the debutanizer? How could opening valve A reduce the apparent tray efficiency of the bottom eight trays from 80 percent to 40 percent?

The temperatures shown in Fig. 49.3 are design temperatures:

- Tray eight temperature = 200°F
- Bottom tray temperature = 300°F
- Reboiler outlet temperature = 400°F
- Tower bottom temperature = 400°F

The tower bottom and reboiler outlet temperatures are both 400°F. This is because all the liquid from tray 1 is supposed to be drawn off to the reboiler. The only liquid in the bottom of the tower is the liquid portion of the reboiler effluent.

But if valve A is opened, and the feed to the reboiler is coming from the bottom of the tower, then the reboiler outlet temperature

must be hotter than the tower bottoms temperature. This temperature difference would be quite small if we had essentially a pure component (like butane) as the tower bottoms product. But with a wide boiling range mixture (like gasoline) as the tower bottoms product, the reboiler outlet temperature would become much hotter than the tower bottom temperature.

As the amount of liquid overflowing the draw pan increases, and the amount of liquid flowing to the reboiler decreases, the tower bottoms temperature will start to approach the tray temperature. For example, with valve A wide open, I had observed the following temperatures:

	Observed	Design
Tray 1	300°F	300°F
Tower bottom	320°F	400°F
Reboiler outlet	400°F	400°F

These temperatures indicated that most of the bottoms gasoline product was coming from the bottom tray of the tower, rather than from the reboiler effluent. Or the once-through thermosyphon reboiler was no longer functioning as the bottom theoretical fractionation stage of my debutanizer.

But so what? So what if I had lost one fractionation stage? I still had eight other trays between the reboiler and the tower feed. So what if I had lost one stage out of nine?

Unfortunately, not all stages are equal. The once-through thermosyphon reboiler functions as a super-fractionation stage. It provides not one-ninth but one-half of the fractionation in the bottom portion of the debutanizer. After all, half of the temperature increase (see Fig. 49.3) between the 200°F tray-8 temperature and the 400°F tower bottoms product temperature occurs across the reboiler, provided that all the liquid from tray 1 flows to the reboiler. The reason the reboiler temperature rise is 100°F is because heat is added to the reboiler. Adding heat to a fractionation stage located below the feed point makes that stage into a super-fractionation stage. If the bottoms product bypasses this super-fractionation reboiler stage, then a large percentage of the tower's potential fractionation capacity is lost.

49.1.1 The Chickens Come Home to Roost

Looking at my computer model of the debutanizer confirmed my worst fears. The butane content of the debutanizer gasoline was 4 percent from the second theoretical stage and 1 percent from the first theoretical stage. Of course, the second theoretical stage of our computer model is tray 1, and the first theoretical stage is the reboiler.

If the sides of the draw-off pan have been left off, then the colder, lighter liquid from tray 1 will bypass the reboiler. This reboiler is doing half of the fractionation work, hence, the overall efficiency of the lower half of the debutanizer falls by one-half. The apparent overall tray efficiency drops from 80 to 40 percent.

My boss generously gave me the opportunity to explain all this to the plant manager. I explained that we had extended the height of the sides of the draw pan by 18 inches. The debutanizer was now back on-line. Gasoline was on specification with 1 percent butane.

"And the new trays?" asked the plant manager. "Were they installed?"

"Not exactly. The new trays have been carefully placed as spares in the refinery warehouse storage," I replied. "They will provide a useful hedge against any future cost inflation in replacement tray charges. They are carried in our accounts as a passive asset."

For years thereafter, the plant manager consistently referred to my anti-inflationary foresight as gross stupidity. His narrow-minded, hostile, and totally unfair comments being based on my supposed negligence in not observing that the sides of the draw pan were missing before the initial start-up in 1969.

49.2 Kettle Reboiler

I could tell almost the same story about the kettle reboiler shown in Fig. 49.2. In this case, we were trying to strip ethane out of a mixture of propane, butane, and gasoline at the Good Hope Refinery in Norco, Louisiana. Valve B had been left open because of fouling and high-pressure drop on the shell side of the kettle reboiler. The kettle reboiler also, like the once-through thermosyphon reboiler discussed above, functions as the bottom theoretical stage of the de-ethanizer. It's a super-fractionation stage because heat is added to the stage. Opening valve B permitted the bottoms product to bypass this super-fractionation stage. We didn't even open the tower. I just had the shell side of the kettle cleaned. We could then run with valve B closed and fractionation efficiency was restored to normal.

49.3 Partial Condenser

Figure 49.4 shows the overhead system of the Coastal Refinery crude distillation tower in Aruba. The island of Aruba is a beautiful country in the Caribbean Sea off the coast of Venezuela. I often take my wife, Liz, on a romantic vacation to this tropical paradise. During one such exotic trip I was assigned by Coastal to find a plan to improve fractionation between naphtha and jet fuel. The problem was that the naphtha contained 20 percent jet fuel. A computer simulation of the crude tower indicated that the apparent tray efficiency of the upper five trays was

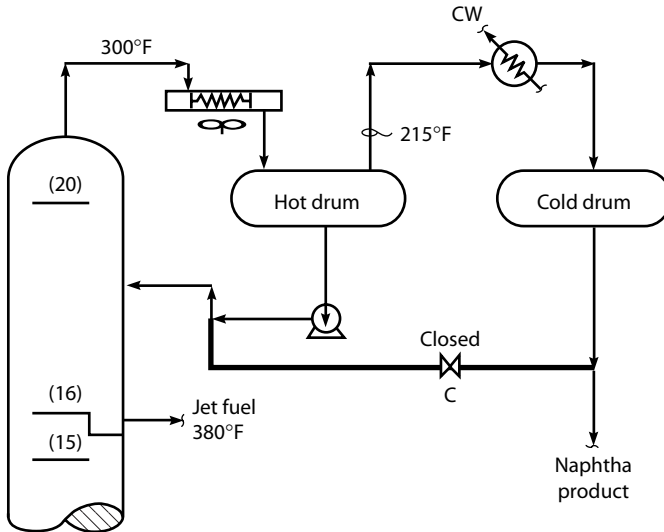


FIGURE 49.4 A partial condenser helps to separate naphtha and jet fuel.

only 30 percent. However, fractionation between naphtha and jet fuel occurs in two regions of the tower shown in Fig. 49.4:

- Trays 16 through 20
- The hot drum, where the reflux liquid is separated from the naphtha vapor product

To discriminate between the fractionation efficiency of the trays and the hot drum, I obtained a sample of reflux from the discharge of the hot drum reflux pump. It contained 25 percent jet fuel and 75 percent naphtha. I compared the analysis of this sample to my computer simulation. The degree of fractionation observed between jet fuel and the reflux indicated a tray efficiency for trays 16 through 20 of 65 percent. This is a normal tray efficiency for this service.

The partial condenser and the hot drum shown in Fig. 49.4 serve as the top theoretical separation stage of this tower. The computer simulation showed that the amount of jet fuel in the reflux should be 25 percent, and the amount of jet fuel in the naphtha product should be 8 percent.

The samples showed that the amount of jet fuel in the reflux was 25 percent, and the amount of jet fuel in the naphtha product was 20 percent. The reason for the high jet fuel content of the naphtha was, I found, that valve C shown in Fig. 49.4 was open halfway. The operators had left valve C open to prevent a high liquid level in the

hot drum. In effect, this ruined the separation efficiency of the top theoretical stage of the tower.

But why would ruining the separation efficiency of one theoretical separation stage degrade the apparent tray efficiency of trays 16 through 20 from 65 to 30 percent? Well, because the top theoretical separation stage represented by the partial condenser is a super-fractionation stage. Heat is removed from that stage. The heat I'm talking about is the duty of the air cooler upstream of the hot drum.

Any time heat is removed from a fractionation stage above a tower's feed point, that stage becomes a super-fractionation stage. To prove my point, note the temperatures shown in Fig. 49.4:

- The ΔT between the jet fuel product draw-off (380°F) and the tower top temperature (300°F) is 80°F.
- The ΔT between the tower top temperature (300°F) and the hot drum temperature (215°F) is 85°F.

For wide boiling range mixtures, such as naphtha and kerosene, the temperature difference across portions of a distillation tower approximately corresponds to the degree of fractionation achieved. The ΔT s indicate that the amount of fractionation work done across the top five trays is about the same amount of fractionation work done in the top theoretical separation stage, which is represented by the partial condenser. The partial condenser consists of the air cooler and the hot drum, shown in Fig. 49.4.

49.3.1 Another Defect

I blocked valve C shown in Fig. 49.4, waited a few minutes, and obtained a sample of product naphtha from the hot drum. Lab analysis showed that the amount of jet fuel in the naphtha product had dropped from 20 to 10 percent. Another victory to report to the plant manager of the Coastal Refinery in Aruba.

Next day, I scheduled a meeting with the plant manager to report on my success. But the plant manager had a different view. His narrow-minded, hostile, and unfair comments were quite offensive.

"Lieberman, I had a check sample run on our naphtha product. We now have 21 percent jet fuel in the naphtha, not 10 percent. You've made fractionation worse, not better."

I rushed out to the unit, suspecting that the operators had reopened valve C (see Fig. 49.4). But it was still shut. What now? Maybe it was best just to accept defeat. But then I looked up at the hot drum, which was located just above valve C. It seemed as if the level shown in the gauge glass was 6 to 10 inches higher than it had been yesterday. I then noticed that there was something odd about the design of the hot drum.

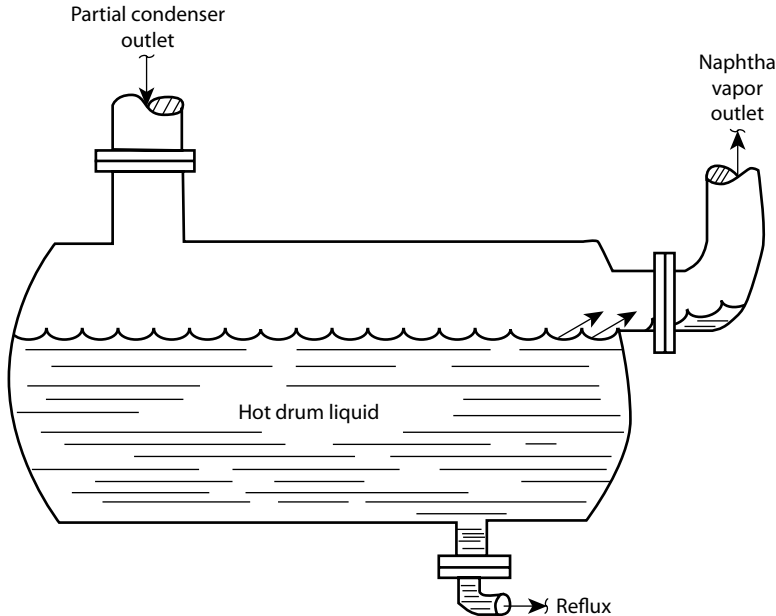


FIGURE 49.5 Liquid carry-over due to a high liquid level.

Referring to Fig. 49.5, note that the vapor outlet nozzle is not on the top of the drum, but on the side. And the liquid level in this drum exactly coincided with the bottom of the vapor outlet nozzle. I suddenly recalled a comment that the panel board operator made yesterday when valve C was closed.

“Señor Lieberman, have you noticed that the hot drum level has increased from 61 to 69 percent?”

I didn’t pay much attention to the comment because the liquid level stabilized at 69 percent. But 69 percent coincided with the bottom of the vapor outlet nozzle. The reflux that had been flowing through valve C into the cold drum was now overflowing into the cold drum through the vapor outlet nozzle.

Overflowing reflux from the hot drum into the cold drum destroyed the efficiency of the super-fractionation stage represented by the partial condenser. To restore this lost fractionation, I had the panel board operator lower the reflux drum level to 60 percent by increasing the tower top reflux rate. This time I waited a few hours before pulling my naphtha product sample. Analysis then showed there was now only 7 percent jet fuel in the naphtha product. Finally, real victory!

Unfortunately, Liz unfairly characterizes this visit to the Coastal Refinery in Aruba as just another business trip. She refuses

to acknowledge that she had been taken to an exotic island paradise for a honeymoon vacation. She has persisted in this narrow-minded view, disregarding the hundred or so trips we have made to the Aruba Refinery, all of which have been, in part, romantic adventures.

49.4 Side Reboilers and Intercoolers

Side or outrigger reboilers are sometimes used to supplement the capacity of the bottom's reboiler. Intercoolers or side slip-stream coolers are sometimes used to supplement the capacity of the overhead condenser.

Side reboilers and intercoolers also function as super-fractionation separation stages. They are just a subcase of once-through thermosyphon reboilers and partial overhead condensers.

CHAPTER 50

Hand Calculations for Distillation Towers

Vapor-Liquid Equilibrium, Absorption, and Stripping Calculations

50.1 Introduction

The top of a distillation tower works like an absorber whereas the bottom of a distillation tower works like a stripper. The upper and lower parts of a tower are the rectification section, which is the upper portion above the feed inlet, and the stripping section, which is the lower portion between the feed inlet and reboiler vapor return nozzle.

In this chapter we will discuss single theoretical stage Bubble-Point and Dew Point calculations and give examples as to how these can be used in our work. Then we'll discuss the use of the Absorption Factor and Stripping Factor chart, which can be used to design distillation towers. In order to use this chart in that way it is necessary to consider a distillation tower as two separate towers. The top of the tower is considered an absorber. The chart is then used in a different way to design the lower part of the distillation tower as a stripper.

Distillation towers were designed using this same chart for nearly half a century before the first computer simulations were ever written. Indeed, the use of computer simulations in design did not really begin until the early 1960s.

As long as we pay attention to the assumptions needed when using this chart, it is still a valuable tool that provides an understanding of how the process works and what to expect when conditions change.

50.2 Bubble Point and Dew Point Calculations

I will demonstrate how to do these calculations, although we have used them elsewhere in this book. But first allow me to point out one of the ways in which you might use them, which is to check on the composition of lab samples. Suppose we take a bomb sample of the vapor and another bomb sample of the liquid from a reflux drum. Working through the Bubble Point and Dew Point calculations will provide a cross-check on the distillation results. If the lab distillation results don't add up to 100%, then something is missing from the analysis. In which case, ask yourself if its light ends that are missing. Perhaps you were sloppy in your sample taking and lost vapors in handling or maybe sampled a liquid that was hot into a bottle instead of taking a bomb sample. Maybe there was residue when the lab performed the distillation so there was thermal cracking during the distillation test. Either way, an analysis that does not add up to 100% indicates something amiss, it could be a problem with your calculation, or with the lab sample, or the lab distillation result.

Norm used to include Bubble Point and Dew Point calculations in his refinery troubleshooting seminars, as a test question for any process engineers who attended (Fig. 50.1). As only about 10% of the engineers could ever answer the question, he has mostly discontinued the use of this in the seminar. But because I like the way the seminar was when I first attended in the 1980s, I think we all deserve to understand this.

50.2.1 Bubble Point

We are going to use a reboiler as an example (see Fig. 50.2).

When vapor is first formed in the reboiler it is in contact with the liquid from which it was formed. We say that the vapor formed while in contact with the liquid from which it was formed is "in equilibrium with" that liquid.

Dalton's law says that the total pressure (P_T) (i.e., in the vapor space in this reboiler) is the sum of the partial pressures of all the components of the vapor.

$$P_T = P_{\text{Butane}} + P_{\text{Propane}}$$

Where P_{Butane} and P_{Propane} are the partial pressures. We could express all of these in psia, or all in mm Hg, or all in atmospheres just as long as we keep the total pressure and partial pressures all in the same units.

Meanwhile Raoult's law allows us to find the partial pressure of a component via its vapor pressure (P_v) and its mol% (X) in the liquid with which the vapor is in equilibrium.

$$P_{\text{Butane}} = P_{\text{V-Butane}} \times X_{\text{Butane}}$$

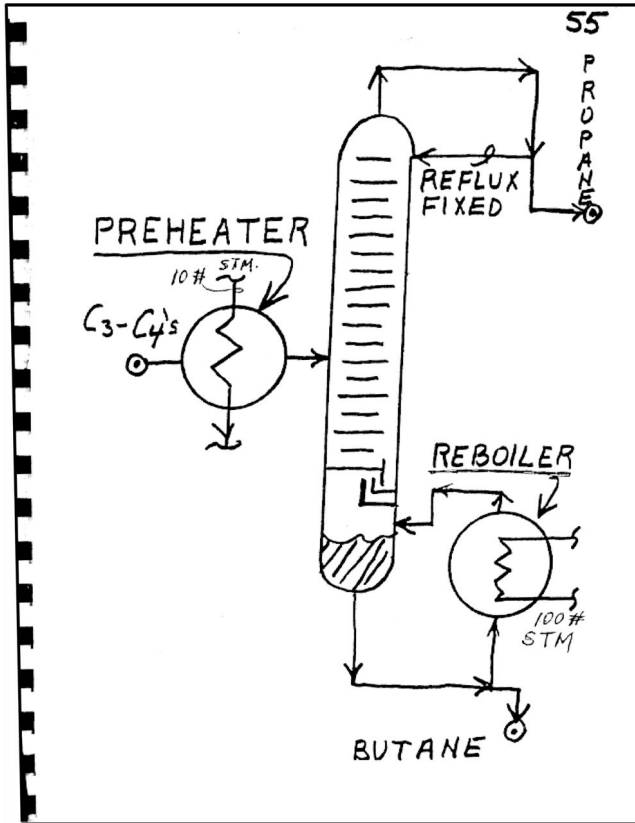


FIGURE 50.1 The bubble point and dew point calculation example, extracted from our seminar manual.

or

$$P_{\text{Propane}} = P_{V,\text{Propane}} \times X_{\text{Propane}}$$

By combining these ideas we can find the mole percentage of each component in the vapor (Y):

$$Y_{\text{Butane}} = \frac{X_{\text{Butane}} \times P_{V,\text{Butane}}}{P_T}$$

or

$$Y_{\text{Propane}} = \frac{X_{\text{Propane}} \times P_{V,\text{Propane}}}{P_T}$$

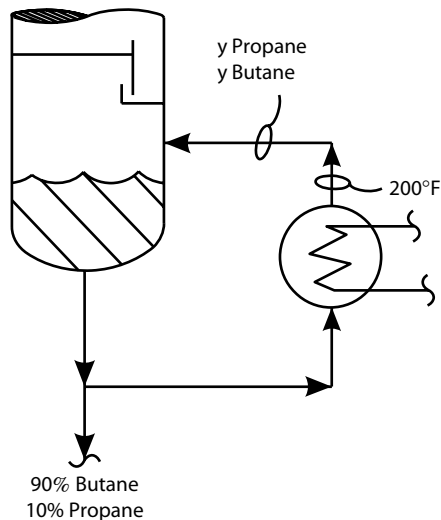


FIGURE 50.2 Using bubble point to calculate vapor composition in a reboiler.

Let's put some numbers to this. We'll take the vapor pressures from a Cox vapor pressure chart (Fig. 50.3).

	Vapor Pressure (P_v) at 200°F in psia	Mol% (x) in Liquid
Butane	200 psia	90%
Propane	600 psia	10%
Partial Pressure (psia)		
Butane	$\frac{90}{100} \times 200 = 180$ psia	
Propane	$\frac{10}{100} \times 600 = 60$ psia	

Hence the total pressure $P_T = 180 + 60 = 240$ psia. And now we can find the mole percent of each component in the vapor as follows:

$$\text{Butane } (180 \div 240) \times 100 = 75\%$$

$$\text{Propane } (60 \div 240) \times 100 = 25\%$$

50.2.2 Dew Point

Looking at Raoult's law again: The partial pressure of a component is given by its vapor pressure multiplied by its mole percentage in the liquid phase, just as we said in the Bubble Point calculation:

$$P_{\text{Butane}} = P_{v,\text{Butane}} \times X_{\text{Butane}}$$

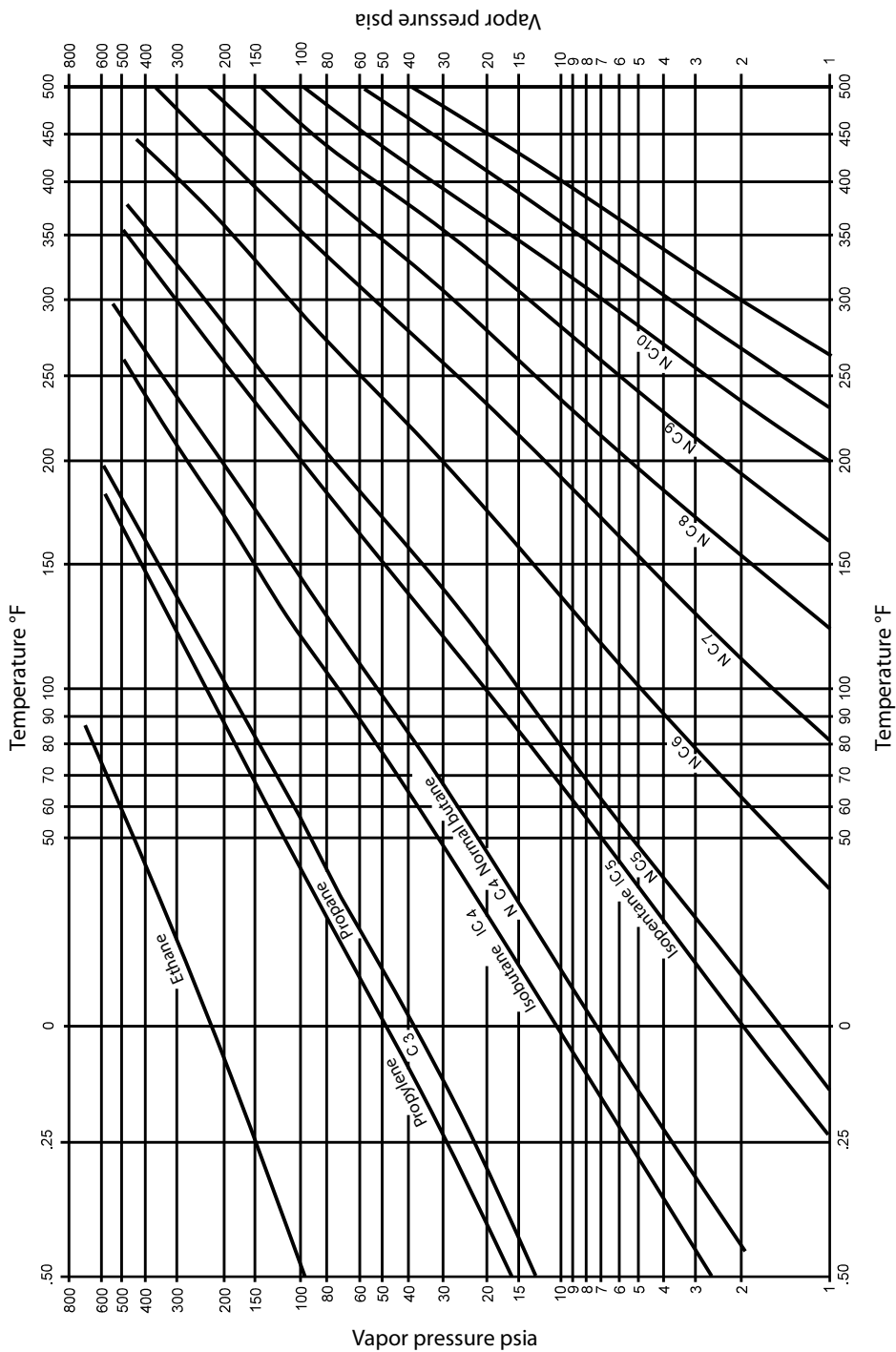


FIGURE 50.3 A Cox vapor pressure chart.

or

$$P_{\text{Propane}} = P_{\text{V.Propane}} \times X_{\text{Propane}}$$

Alternatively we could write this as:

$$= \frac{P_{\text{Butane}}}{P_{\text{V.Butane}}} = X_{\text{Butane}}$$

or

$$= \frac{P_{\text{Propane}}}{P_{\text{V.Propane}}} = X_{\text{Propane}}$$

Whereas for the Bubble Point calculation we calculated the vapor composition in equilibrium with a liquid, for the Dew Point calculation we are given the vapor composition and will calculate the composition of the liquid that is in equilibrium with the vapor.

Suppose we did not know the liquid composition for the bottom product as was given in the Bubble Point calculation in the previous section, but instead we did have the composition of the vapor at the operating temperature so that the partial pressure of each component was also easy to calculate. We could then find the composition of the liquid by calculating the mol percentage of each component in the liquid that was in equilibrium with the vapor.

For example:

$$X_{\text{Butane}} = \frac{P_{\text{Butane}}}{P_{\text{V.Butane}}} \rightarrow \frac{180 \text{ psia}}{200 \text{ psia}} \times 100 = 90\%$$

or

$$X_{\text{Propane}} = \frac{P_{\text{Propane}}}{P_{\text{V.Propane}}} \rightarrow \frac{(60 \text{ psia})}{(600 \text{ psia})} \times 100 = 10\%$$

But although using the exact same set of numbers as for the Bubble Point calculation may help demonstrate this Dew Point calculation, let's try a different example to demonstrate the Dew Point calculation.

Let's look at the reflux drum of that same distillation tower that we used for the Bubble Point calculation (see Fig. 50.4). Again the vapor coming off the reflux drum is in equilibrium with the liquid in the reflux drum. Using our Cox vapor pressure chart (Fig. 50.3), we find the following.

In this case we do not have the total pressure (P_T) in the vapor space above the liquid in the reflux drum, but we still want to find the composition of the liquid in the reflux drum that is in equilibrium with that vapor.

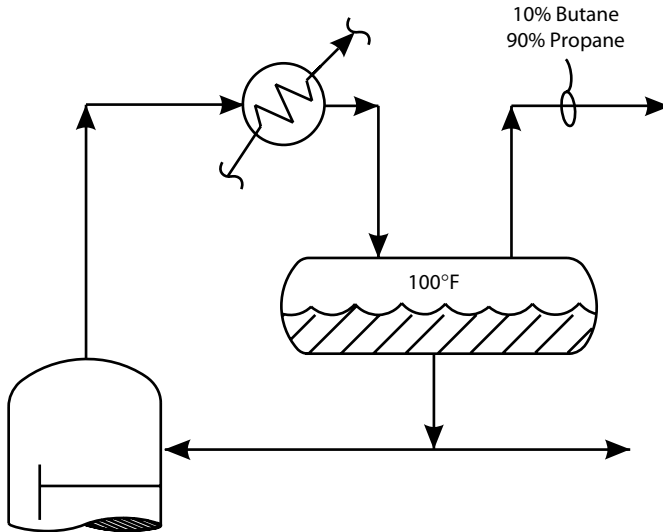


FIGURE 50.4 Overhead of a $C_3 - C_4$ splitter.

Traditional calculation methods allow us two choices. At this point we have two options.

	Vapor Pressure (P_v) at 100°F in psia	Mol% (y) in Vapor
Butane	50 psia	10%
Propane	180 psia	90%

50.2.2.1 Option A

Make a guess at the total pressure (P_T), then use the relationship from Dalton's law to find the partial pressures of each component, as in:

$$\text{Partial Pressure} = \text{Mol\% in Vapor (Y)} \times \text{Total Pressure (P}_T\text{)}$$

Then having found the partial pressure for each component, we could use the relationship from Raoult's law to find the mol% of each component in the liquid which is in equilibrium with the vapor, as in:

$$\text{Mol\% in Liquid (X}_i\text{)} = \frac{\text{Partial Pressure } P_i}{\text{Vapor Pressure } P_{v,i}}$$

Where i in this case would be either butane or propane.

Having then calculated mol% in the liquid for each component, if these do not add up to 100% when added together, one must start

all over again at the beginning of Option A by making a revised guess of the total pressure.

This kind of calculation is known as *iterative*. One would keep going through this procedure making new guesses at the total pressure until such time as the mol% for each component, when all added together, did reach 100%, no more and no less than 100%. Each time we go through a series of calculations like this we call it an *iteration*. This method could also be referred to as an iterative method, iterating on Total Pressure.

50.2.2.2 Option B

The traditional alternate would be to use the relationship with the Equilibrium constant, also known as the equilibrium ratio, K_i .

$$y_i = K_i \times x_i$$

Where i in this case would be either butane or propane.

$$K_i = \frac{y_i}{x_i}$$

With

$$y_i = \frac{\text{Partial Pressure } (P_i)}{\text{Total Pressure } (P_T)}$$

$$x_i = \frac{\text{Partial Pressure } (P_i)}{\text{Vapor Pressure } (P_{v,i})}$$

In order to use this method we need an extra chart to provide K_i for different values of temperature. Such a K_i versus T chart is given in Chap. 11 of *Chemical Engineering: Vol. II*, 3rd ed. by Coulson and Richardson.⁴ Actually this chart in the third edition is in SI units, so we would need to convert the temperature from degrees Fahrenheit to absolute temperature (i.e., Kelvin or K) in the SI system to use it. This is done as follows:

$$100^\circ\text{F} = \frac{(100 - 32)}{180} \times 100 \rightarrow 37.8^\circ\text{Centigrade } (^\circ\text{C})$$

$$37.8^\circ\text{C} \rightarrow (37.8 + 273) \rightarrow 311\text{K}$$

But here at last is a nontraditional and third option for this Dew Point calculation. It is a direct calculation using the vapor pressure chart.

50.2.2.3 Option C

Refer again to Fig. 50.4, the reflux drum of the C_3C_4 splitter with 10 mol% Butane and 90 mol% Propane in the vapor, and the reflux drum temperature at 100°F .

	Vapor Pressure (P_v)	Mol% (y) in Vapor
Butane	50 psia	10%
Propane	180 psia	90%
$y_i/P_{v,i}$		
Butane	Mol% ÷ Vapor Pressure $\frac{10}{50} = 0.2$	
Propane	Mol% ÷ Vapor Pressure $\frac{90}{180} = 0.5$	

In this case i is either butane or propane. The next step is to add together all the values of $y_i \div P_{v,i}$ for each component:

$$\Sigma \left(\frac{y_i}{P_{v,i}} \right) \rightarrow 0.2 + 0.5 \rightarrow 0.7$$

The last step is to calculate for the mol% for each component in the liquid (X_i) from:

$$X_i = \left(\frac{Y_i}{P_{v,i}} \times \frac{1}{\Sigma \left(\frac{Y_i}{P_{v,i}} \right)} \right) \times 100$$

$$X_{\text{Butane}} = \left(0.2 \times \frac{1}{0.7} \right) \times 100 \rightarrow 28.6 \text{ mol\% Butane in reflux drum liquid}$$

$$X_{\text{Propane}} = \left(0.5 \times \frac{1}{0.7} \right) \times 100 \rightarrow 71.4 \text{ mol\% Propane in reflux drum liquid}$$

We call this Option C direct calculation method, “Norm’s method.” It’s not found in any of our textbooks or notes, but it works. If you don’t believe us, check the validity as follows, using the values for X that we just found. From Raoult’s law:

Partial Pressure = Vapor Pressure \times Mol% in Liquid

$$P_{\text{Butane}} = 50 \times \frac{28.6}{100} = 14.3 \text{ psia}$$

$$P_{\text{Propane}} = 180 \times \frac{71.4}{100} = 128.5 \text{ psia}$$

Thus total pressure (P_T) in reflux drum via “Norm’s method” is:

$$P_T = 14.3 + 128.5 = 142.8 \text{ psia}$$

Now we can check Option C, “Norm’s method,” using our Bubble Point calculation method together with the total pressure that we have just found ($P_T = 142.8$ psia) and the liquid composition that “Norm’s method” gave us for the reflux drum, which was 28.6% Butane and 71.4% Propane, to see if we can arrive at the vapor composition as shown in Fig. 50.4.

Thus via Bubble Point calculation:

	Vapor Pressure (P_v)	Mol% in Liquid (x)	Partial Pressure (psia)
Butane	50	28.6	$\frac{28.6}{100} \times 50 = 14.3$
Propane	180	71.4	$\frac{71.4}{100} \times 180 = 128.5$

$$\text{Total Pressure} = 14.3 + 128.5 \rightarrow 142.8 \text{ psia}$$

Thus the mol% (Y) of each component in the reflux drum vapor is:

$$\text{Butane } \frac{14.3}{142.8} \times 100 = 10\%$$

$$\text{Propane } \frac{128.5}{142.8} \times 100 = 90\%$$

We were satisfied with this and hope you will find Option C a useful tool.

50.3 The Absorption Factor or Stripping Factor Chart

The Absorption factor or Stripping factor chart, as it is sometimes known, is shown in Fig. 50.5. It provides simple calculation methods for hydrocarbons when considering either an Absorber tower or a Stripper tower in hydrocarbon service, but for combination Absorber-Stripper towers the calculation procedures become iterative and a lot more complex. So for a combination Absorber-Stripper tower it is best not to attempt to use this chart. It is actually better to resort to a computer simulation. In fact, as a point of historical interest, Norman tells me that it was when the idea of combined Absorber-Stripper towers was first invented that brought about the use of computer simulations so as to handle all the intricate and cumbersome calculations needed to design them.

But let’s go back to the use of this chart for the separate Absorber tower or separate Stripper tower. First, the Absorption Factor, A , is defined as:

$$A = \frac{L}{V_x K_i}$$

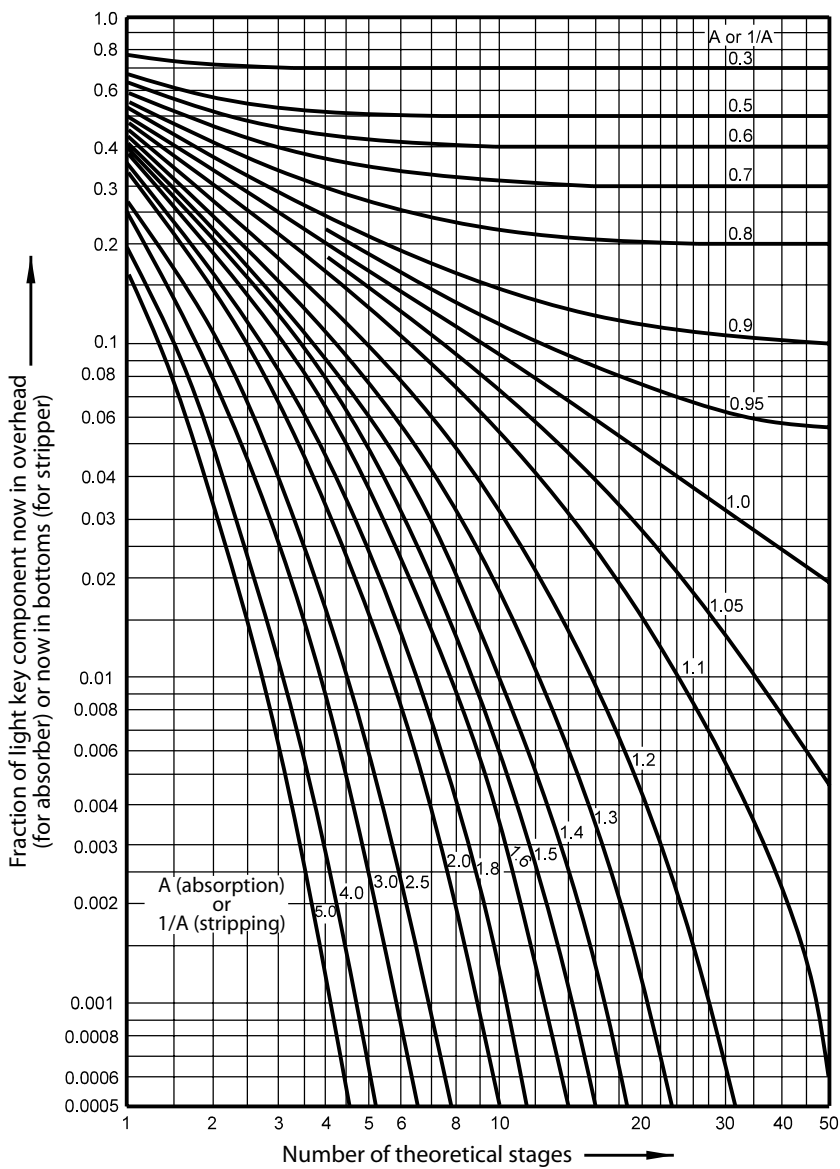


FIGURE 50.5 Absorption factor or stripping factor chart.

where A = absorption factor

L = moles of liquid

V = moles of vapor

K_i = equilibrium constant or equilibrium ratio, as previously defined in bubble point and dew point calculations as:

$$K_i = \frac{y_i}{x_i}$$

where

$$y_i = \frac{P_i}{P_T}$$

And

$$x_i = \frac{P_i}{P_{v,i}}$$

or as:

$$K_i = \frac{P_i}{P_T} \times \frac{P_{v,i}}{P_i}$$

where P_i = partial pressure of component i

P_T = total pressure

$P_{v,i}$ = vapor pressure of component i

$$\text{Then } K_i = \frac{P_{v,i}}{P_T}$$

Hence the absorption factor is also defined as:

$$A = \frac{L}{V} \times \frac{P_T}{P_{v,i}}$$

As the action of stripping is taken to be effectively the reverse of absorption, the stripping factor for which we write S is defined as the reciprocal of the absorption factor.

Thus:

$$S = \frac{1}{A}$$

or

$$S = \frac{VK}{L}$$

or

$$S = \frac{V}{L} \times \frac{P_{v,i}}{P_T}$$

The chart assumes that the absorption factor, A , will remain fairly constant over the range of theoretical stages. The horizontal

axis describes the number of theoretical stages we have assumed 50 percent efficiency for the trays in the tower, which means that whatever the number of trays in the tower, we then have half that number as the number of theoretical stages.

The vertical axis refers to the amount rejected or the amount lost. For an absorber tower this is the amount of the light key component not absorbed in the lean oil, but when the chart is used for a stripper tower, this is the amount not stripped out. In reference to this chart the term “amount” that I am using refers to the fraction and not the percent; this just means that the sum of all the fractions of the light key component present in the system add up to 1. Whereas if we refer to the percent, then the sum of each of the percentages of the light key component present would add up to 100. The vertical axis also assumes zero light key component in the lean oil, which is often not the case.

A more classical definition of the vertical axis of this chart can be found in either *Mass Transfer Operations* by R. E. Treybal or *Separation Processes* by C. J. King,^{5,6} but be warned, neither of these books is for the faint of heart, especially in the sections where absorption or stripping are concerned. So if you choose to look up these references, please don't get despondent as some of the verbiage (or in places, lack of verbiage) would break the heart of the most aspiring and ardent student.

Let us now clarify the use of this handy chart with some examples.

50.3.1 Absorption or Stripping Factor Chart for Absorption

A wet gas stream containing propane as the light key component from a cat cracker (FCC) is being fed to an absorber (see Fig. 50.6).

The absorber tower has 20 trays, which for this calculation we will assume to be 50 percent efficient so we can say that this tower has 10 theoretical stages.

The lean oil rate is 2000 barrels per day, and with existing current operation there is 90 percent recovery of the propane to the rich oil (i.e., 10 percent loss of propane to the dry gas). We will assume zero light key component in the lean oil (i.e., we are assuming there is no propane in the lean oil).

$$\text{Lean Oil Rate} = 2000 \text{ B/D}$$

$$90\% \text{ Recovery Propane to Rich Oil} = 10\% \text{ Loss Propane to Dry Gas}$$

$$= \frac{10}{100} \text{ Loss Propane to Dry Gas}$$

$$= \frac{0.1}{1} \text{ Loss Propane to Dry Gas}$$

$$20 \text{ trays at } 50\% \text{ efficiency} = 10 \text{ Theoretical Stages}$$

We now plot this existing operation on the chart to find A (the absorption factor). See Fig. 50.7. From the chart we find:

$$A = 0.98$$

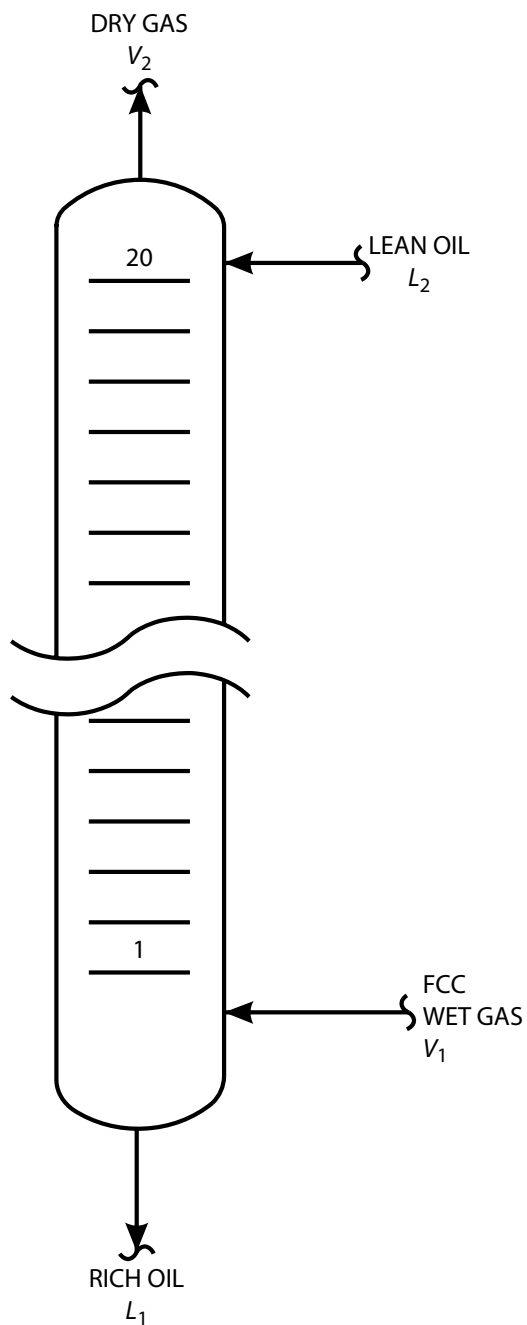


FIGURE 50.6 An FCC wet gas absorber tower.

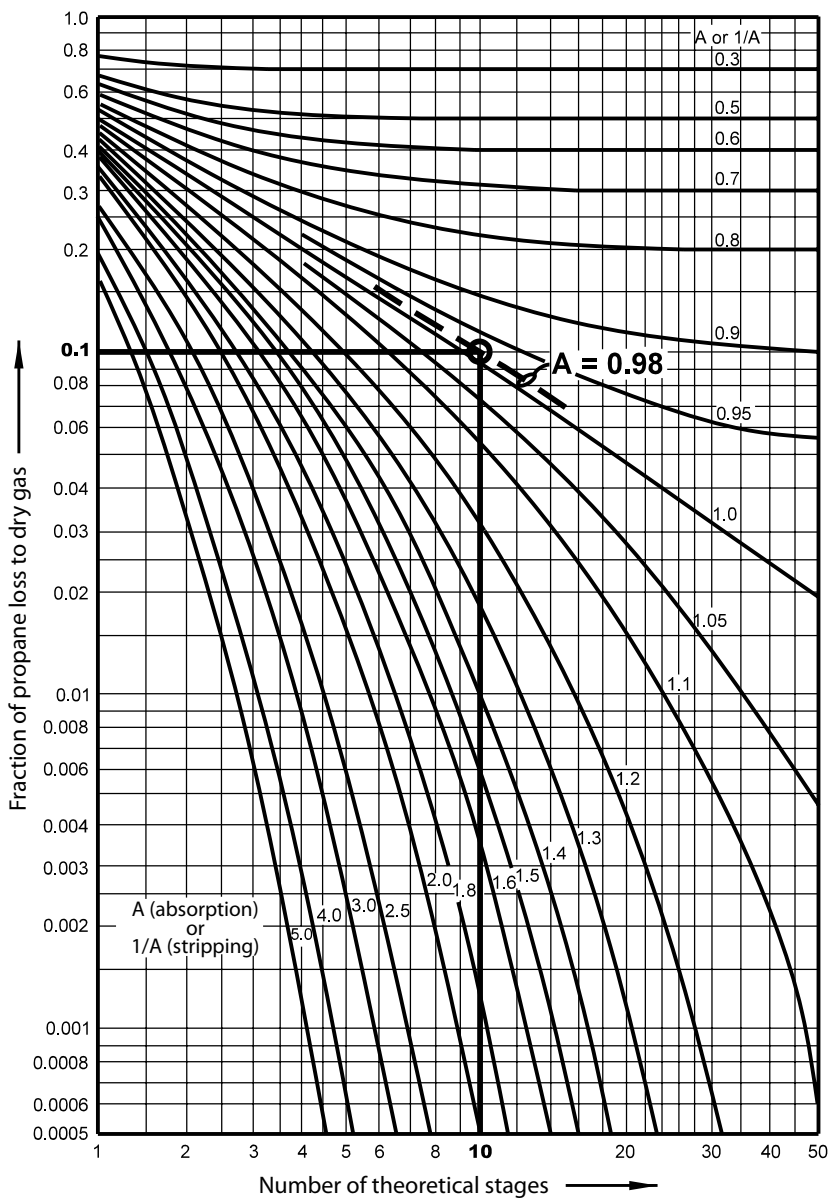


FIGURE 50.7 Absorption plot for an FCC absorber with 10 theoretical stages and 10% loss of propane from feed to dry gas.

Management wants to improve recovery of the Propane, so operations decides to increase the lean oil rate. If the lean oil rate is now doubled, and we assume that the vapor rate (V) and equilibrium constant (K) remain the same at the increased lean oil rate as they were at the original lean oil rate, then we can simply multiply the absorption factor by 2 so as to obtain the new absorption factor at the new lean oil rate of 4000 barrels per day, as follows:

Old lean oil rate 2000 B/D

$$A_{\text{old}} = \frac{2000}{V \times K} = 0.98$$

New lean oil rate 4000 B/D

$$2 \times A_{\text{old}} = 2 \times \frac{2000}{V \times K}$$

Thus

$$2 \times 0.98 = \frac{4000}{V \times K}$$

Therefore

$$A_{\text{at 4000 B/D}} = 1.96$$

Hence the absorption factor A at 4000 B/D is also doubled, but the number of theoretical stages is still 10. Again we refer to the chart and this time plot the condition at the new lean oil rate of 4000 barrels per day. (See Fig. 50.8.)

Lean Oil Rate 4000 B/D

Theoretical Stages = 10

$$A_{4000 \text{ B/D}} = 1.96$$

Assume no propane in the lean oil. From the chart (Fig. 50.8):

$$\begin{aligned} \text{Loss of Propane to Dry Gas} &= \frac{0.0007}{1} \\ &= \frac{0.07}{100} \\ &= 0.1\% \end{aligned}$$

Therefore,

Percent Recovery of Propane to Rich Oil

$$= (100 - 0.1)$$

$$= 99.9\% \text{ Recovery of Propane to Rich Oil}$$

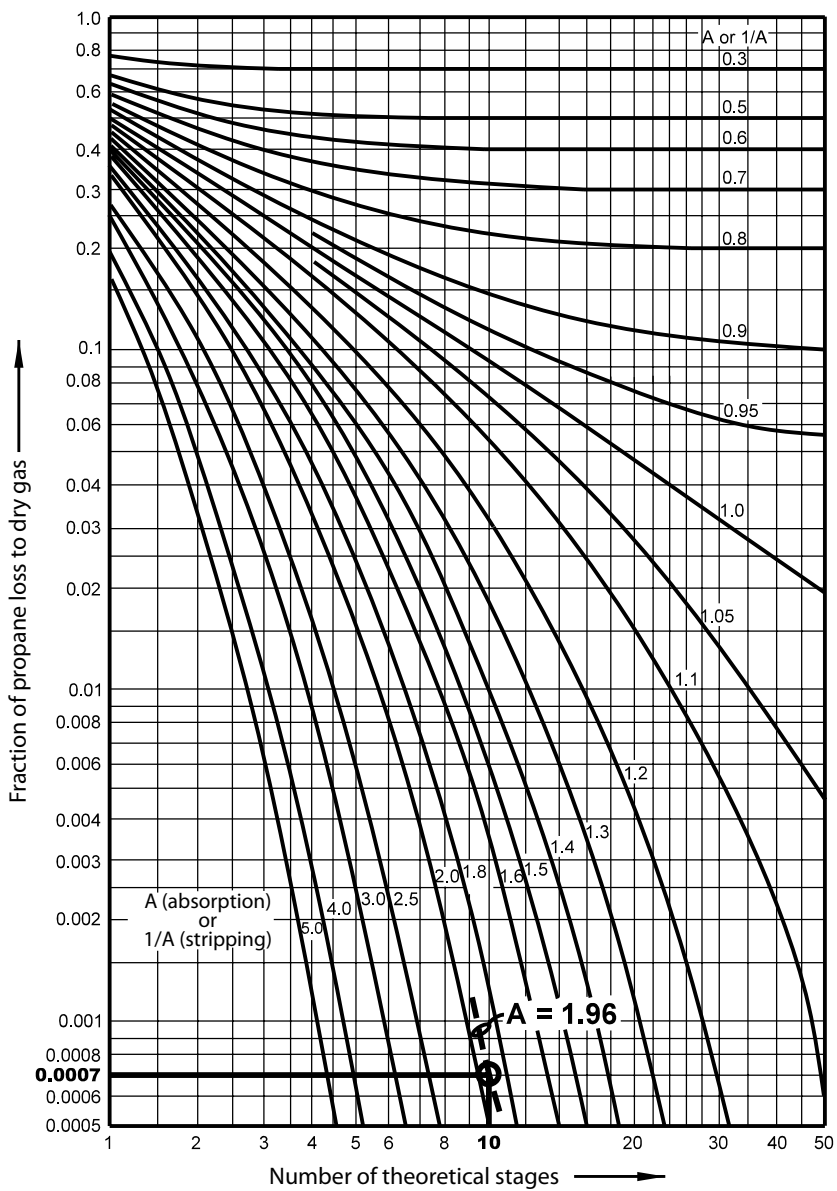


FIGURE 50.8 Absorption plot for an FCC absorber with 10 theoretical stages and $A = 1.96$.

This represents an approximate 10 percent increase in recovery of propane to rich oil by doubling the lean oil rate.

You may note that in order to use this chart we have had to sketch in a line to represent the appropriate line for the value of the absorption factor between existing lines for A on the chart. Also, we should be cautious about assuming that there is none of the light key component (in this case Propane) in the lean oil, as this is not always true.

If you find that the light key component is present in the lean oil, then the method described above may not be valid, and you should refer to one of the classical reference texts such as *Mass Transfer Operations* by Treybal or *Separation Processes* by King^{5,6} for guidance as to how to proceed with the calculation. Note that the higher the recovery of the light key component, the more critical it becomes that there should be no light key component in the lean oil, otherwise the method we show here may not produce correct results.

50.3.2 Stripping Factor or Absorption Factor Chart for Stripping

50.3.2.1 Case I: Increasing the Number of Trays

A de-ethanizer tower has 10 trays, we will assume the trays have 50 percent efficiency in this case, hence, there will be five theoretical stages. Ninety percent of the Ethane is being stripped out of the Naphtha stream, thus 10 percent of the Ethane fed to the tower in the Naphtha stream is left in the bottoms product (Fig. 50.9).

Summarizing:

$$\begin{aligned} 10 \text{ Trays at } 50\% \text{ Efficiency} &= 5 \text{ Theoretical Stages} \\ 90\% \text{ Rejection of Ethane Overhead (V}_2\text{)} \end{aligned}$$

Therefore

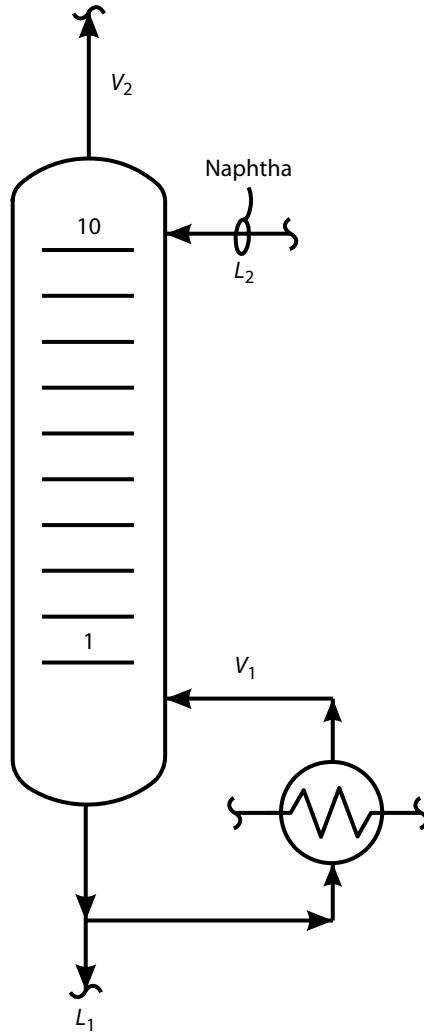
$$10\% \text{ Loss of Ethane in Bottoms Product (L}_1\text{)}$$

$$\begin{aligned} &= \frac{10}{100} \text{ Loss of Ethane in Bottoms} \\ &= \frac{0.1}{1} \text{ Loss of Ethane in Bottoms} \end{aligned}$$

Refer to Fig. 50.10. We will now use the lines that we used in the previous example for absorption as A for the reciprocal of A, that is, we will now take the alternate use of these lines as

$$\frac{1}{A} = \text{Stripping Factor S}$$

FIGURE 50.9 Stripping in a de-ethanizer.



where $S = \frac{VK}{L}$

From the chart,

$$\frac{1}{A} = S = 1.2$$

If we now double the number of trays so that there are now 20 trays with 50 percent efficiency, then the number of theoretical stages will now be 10, but the stripping factor S will remain the same: 1.2.

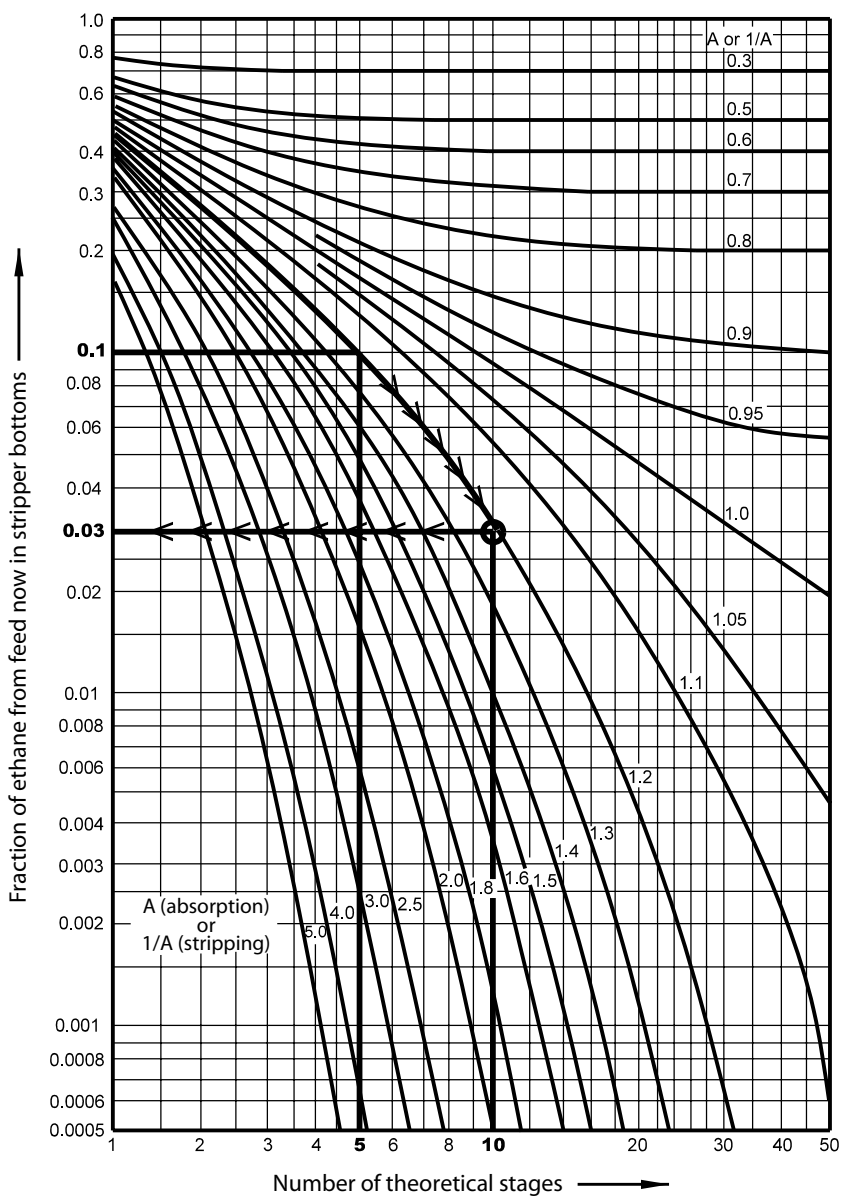


FIGURE 50.10 Stripper with $S = 1/A = 1.2$ has 5 theoretical stages, then doubled theoretical stages.

Summarizing:

Old Number of Trays = 10

Efficiency of Old Trays = 50%

Number of Theoretical Stages = 5

Ethane remaining in bottoms = 10% of Ethane in feed

Stripping factor = 1.2

New Situation:

New Number of Trays = 20

Efficiency of Trays = 50%

Number of Theoretical Stages = 10

Stripping factor $S = 1.2$

To find the new amount of Ethane remaining in the bottoms now that we have doubled the number of theoretical stages, we need to refer to the chart again. See Fig. 50.10. Follow the curved line for " $1/A = 1.2$ " to the point where it intersects the vertical line for 10 theoretical stages.

Next read off the fraction of Ethane that now remains in the bottoms from the vertical axis on the left to find, that is, percent of ethane from the feed now remaining in the bottoms. That is equivalent to 97 percent of ethane in the naphtha feed now stripped out.

Summarizing, we now have:

Theoretical Stages = 10

$S = 1.2$

$$\begin{aligned}\text{Light key component Ethane from feed now in bottoms} &= \frac{0.03}{1} \\ &= \frac{3}{100} \\ &= 3\%\end{aligned}$$

Therefore,

$$\begin{aligned}\text{Light key component Ethane from feed now stripped out} &= \frac{(100 - 3)}{(100)} \\ &= 97\%\end{aligned}$$

50.3.2.2 Case II: Increasing the Rate of Stripping Vapor

In this case the de-ethanizer shown in Fig. 50.9 is stripping out 80 percent of the ethane from the naphtha feed stream. Thus there is 20 percent of the ethane from the feed left in the bottoms. There are

10 trays with 50 percent efficiency, therefore we have five theoretical stages.

Summarizing:

$$\text{Number of Trays} = 10$$

$$\text{Tray Efficiency} = 50\%$$

$$\text{Number of Theoretical Stages} = 5$$

$$\text{Light key component Ethane stripped out} = 80\%$$

$$= \frac{80}{100}$$

$$= \frac{0.8}{1}$$

$$\text{Light key component ethane remaining in bottoms} = \frac{100 - 80}{100}$$

$$= 20\%$$

$$= \frac{0.2}{1}$$

Therefore,

$$\text{Value for the vertical axis} = 0.2$$

We now refer to the chart shown in Fig. 50.11. Plot 0.2 (feed ethane remaining in bottoms) on the vertical axis and 5 (theoretical stages) on the horizontal axis. The intersection on the chart gives a stripping factor of 0.92. Thus:

$$\text{Ethane from feed now in bottoms} = 0.2$$

$$\text{Theoretical Stages} = 5$$

Then from the chart:

$$\frac{1}{A} = S = 0.92$$

Suppose we now double the vapor rate by doubling the reboiler duty. There will still be five theoretical stages. But the stripping factor is now doubled because we are assuming that the naphtha feed rate and bottoms product rate remain constant. Thus:

$$\frac{1}{A} = S = \frac{V}{L} \times K$$

If

$$V_{\text{New}} = 2 \times V_{\text{Old}}$$

and

$$L_{\text{New}} = L_{\text{Old}}$$

K = Equilibrium constant and remains unchanged because the pressure and temperature are constant. Therefore

$$\begin{aligned}S_{\text{New}} &= 2 \times S_{\text{Old}} \\S_{\text{New}} &= 2 \times 0.92 \\&= 1.84\end{aligned}$$

Again summarizing:

$$\text{Theoretical Stages} = 5$$

Ethane as light key from feed now in bottoms at old vapor rate = $\frac{0.2}{1}$

$$\text{Stripping factor } S \text{ at old vapor rate} = 0.92$$

New situation:

$$\begin{aligned}\text{Theoretical Stages} &= 5 \\ \text{New Vapor Rate} &= 2 \times \text{Old Vapor Rate}\end{aligned}$$

Therefore

$$\text{New Stripping Factor } S = 1.84$$

Therefore

$$\frac{1}{A} = 1.84$$

Referring to Fig. 50.11, follow the line for five theoretical stages to the point where it meets the line for a stripping factor of 1.84. Next read the new value for the ethane from feed remaining in the bottoms as 0.02. Thus for this new situation with double the vapor rate:

$$\begin{aligned}\text{Theoretical Stages} &= 5 \\ \text{Stripping Factor } S &= 1.84\end{aligned}$$

From the chart,

Ethane as light key from feed now in bottoms at new double vapor rate

$$\begin{aligned}&= \frac{0.02}{1} \\&= \frac{2}{100} \\&= 2\%\end{aligned}$$

Thus

$$\begin{aligned}\text{Ethane from feed now stripped out} &= \frac{100 - 2}{100} \\&= \frac{98}{100} \\&= 98\%\end{aligned}$$

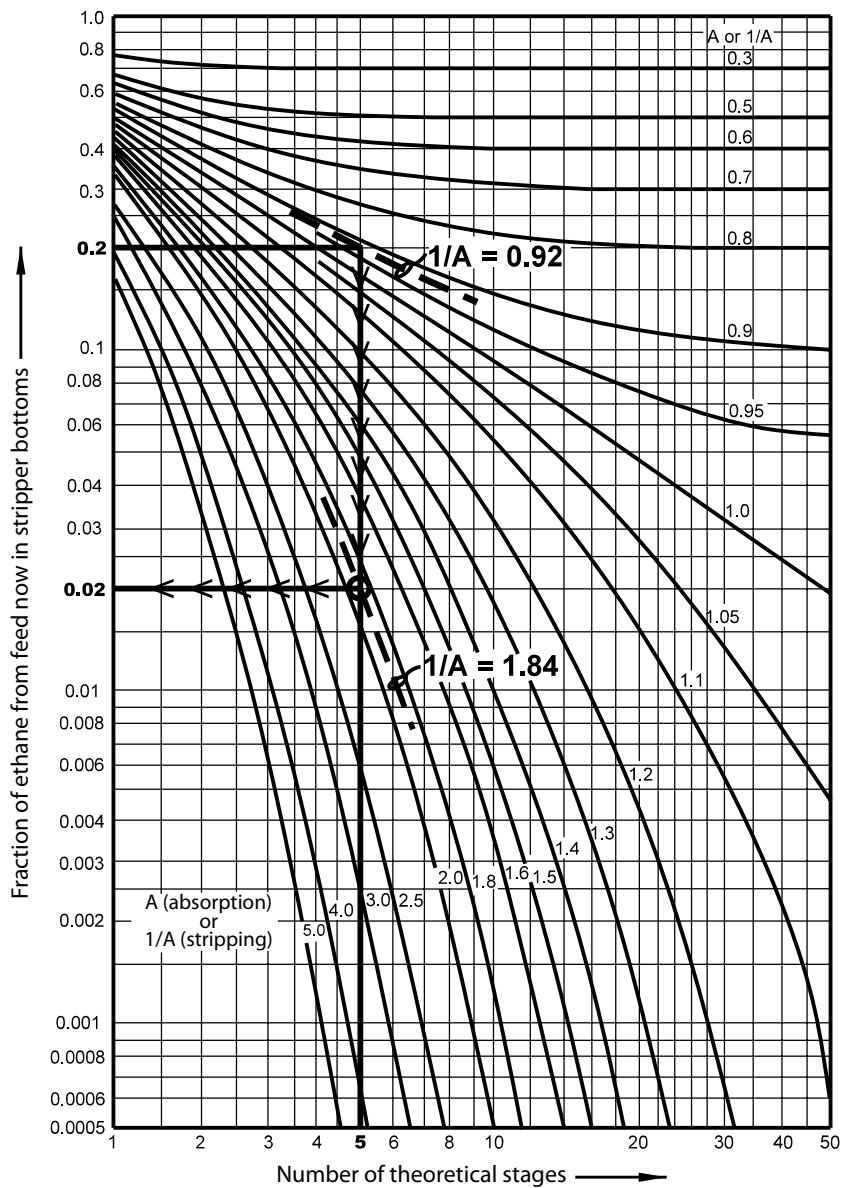


FIGURE 50.11 Stripper with 5 theoretical stages with doubled stripping vapor rate.

Hence by doubling the vapor rate on this particular tower we increased the amount of ethane stripped out from the feed from 80 percent to 98 percent, so we increased the amount stripped out 18 percent based on the amount of ethane in the feed.

$$\frac{98-80}{100} \rightarrow \frac{18}{100}$$
$$= 180\%$$

50.4 Conclusion

As with all calculation methods, remember to pay attention to the assumptions made and make sure that they are valid for the process with which you are working.

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CHAPTER 51

Computer Modeling and Control

When I first went to work in the process industry in 1964, computer control was in its infancy. The use of computer modeling of distillation towers was just beginning. I believe I was the first person in the oil refining industry to model a crude distillation tower with:

- Multiple pumparounds
- Side-stream steam strippers
- Multiple side-product draw-offs
- Two-stage overhead system

In 1968, I was the technical support engineer on the crude distillation unit at the Amoco Refinery in Whiting, Indiana. My job was to implement a new computer control program for the automated operation of the unit. Basically, I was to convince the operators that automated, direct, on-line computer control could produce more consistent product specifications than an experienced panel board operator. One of the factors that led to my failure in this assignment was that in 1968 I was not sure if this was true. Now, in 2014, I'm still not convinced that direct computer control of refinery processes is always the best approach.

The situation regarding computer modeling is somewhat more straightforward. No engineering calculation can be better than the assumptions used as a basis to carry out the calculations.

51.1 Modeling a Propane-Propylene Splitter

I had been hired to review the design for a new 12,000 BSD propane-propylene splitter for a chemical plant in Convent, Louisiana. The tower was intended to be a replacement for an existing 10,000 BSD propylene-propane splitter. The incremental capacity of 2000 BSD would economically justify the \$3,000,000 investment for the new splitter.

I asked the operators why the existing tower was limited to its current 10,000 BSD capacity. They responded by noting that above 10,000 BSD of feed, the concentration of propane in the propylene overhead product (see Fig. 51.1) would exceed the 2 percent specification.

Dear reader, note that I'm already in trouble. My client had asked me to review the design for the new tower. No one had asked me to review the operation of the existing splitter. However, if anybody had asked what I was doing, I had a ready response: I'm gathering data to verify the accuracy of the computer model used in the design of the new tower. I'm collecting operating data to check the following parameters *assumed* in the computer design of the new splitter:

- Tray efficiency
- Relative volatility for the propylene-propane system (i.e., the "equation of state")

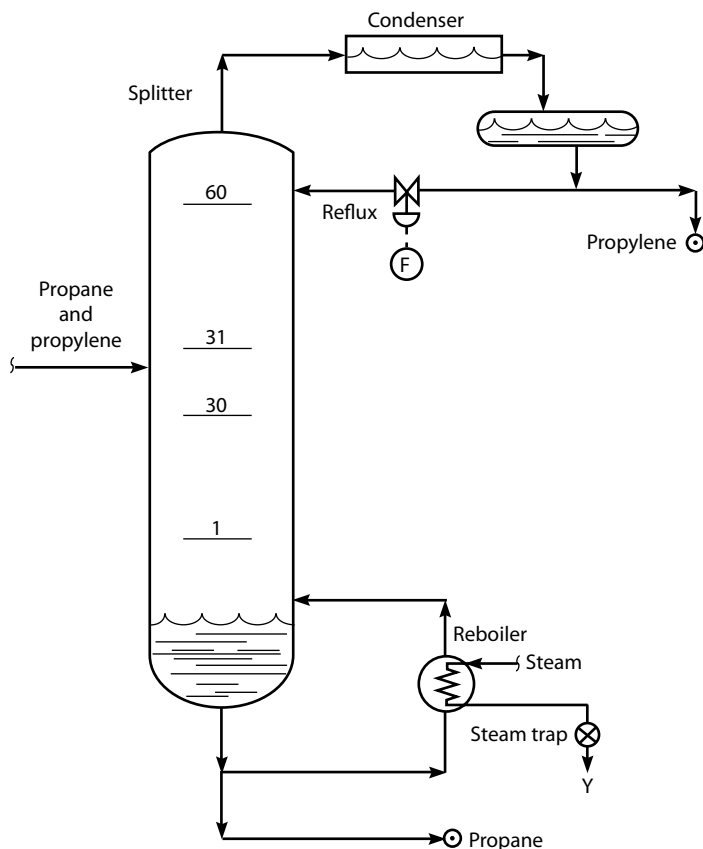


FIGURE 51.1 A propane-propylene splitter limited by computer control.

I asked the operators to increase the splitter feed rate by 500 BSD to 10,500 BSD. When they complied, the concentration of propane in the overhead propylene product increased from 2 to 3 percent.

"How about increasing the reflux rate from 20,000 BSD to 21,000 BSD? We have to keep the reflux ratio constant as we increase the feed rate," I explained.

"Can't be done, Norm," responded the panel board operator. "The 20,000 BSD reflux rate is our maximum. The reflux flow control valve is programmed to avoid excessive reflux rates."

"Excessive? What's excessive about 21,000 BSD of reflux? The reflux flow control valve is only 45 percent open. Let's try for 21,000 BSD," I persisted.

"Now, Norm, it's part of the computer control logic not to exceed 20,000 BSD of reflux. It's part of the Black Box. These sorts of parameter limits are imbedded in the computer control logic," explained the board operator. "Frank Skorski spent a lot of time on the computer control of this splitter. I imagine Frank knew a hell of a lot more about computer control than you. He was the computer control engineer here in Convent for 12 years. Norm, I don't think you should be second-guessing Mr. Skorski."

"Let's call Mr. Skorski and get his okay to raise the reflux rate," I suggested.

"No, we can't," replied the operator.

"Why?"

"Because Frank Skorski has been dead for three years."

It required three hours to get the artificial limit on the reflux rate cancelled. Frank Skorski's 20,000 BSD reflux limit had been well founded. It reflected a lack of reboiler duty (see Fig. 51.1) needed to generate the reflux. The splitter was short of reboiler duty because the condensate steam trap was undersized. A larger steam trap had been installed last year. However, Mr. Skorski had failed to return from the spirit world to remove the reflux limit.

Gradually we increased the tower feed rate to 12,000 BSD, and the reflux rate to 24,000 BSD. The online gas chromatograph stayed at the desired 2 percent propane in the propylene overhead product.

51.1.1 Predicting Tray Efficiency vs. Relative Volatility

Why had my client decided to build a new splitter when the existing tower had the required capacity?

The process design engineer for the project informed me that the existing tower did not have sufficient tray hydraulic capacity to handle the required feed rate of 12,000 BSD and the required reflux rate of 28,000 BSD. But what was the basis for the 28,000 BSD of reflux? Hadn't I just demonstrated that a 24,000 BSD reflux rate was sufficient? True, the tower would flood at 28,000 BSD reflux. True, the computer model indicated that 28,000 BSD of reflux was needed. But was the computer model used for this propane-propylene splitter correct?

51.1.2 The Input Data Problem

When we model any piece of process equipment, some of the data we supply to the computer to create the model are derived from the observed performance of existing equipment in similar service. For instance:

- Heat exchangers—Fouling factors
- Compressors—Adiabatic compression efficiency
- Centrifugal pumps—Clearance between impeller, pump case and wear ring
- Distillation towers—Tray efficiency and relative volatility

The relative volatility is a measure of the ease of separation between the light and heavy components. For ideal components it is the ratio of the vapor pressure of the more volatile component divided by the vapor pressure of the less volatile component. For nonideal components, the relative volatility is calculated from the “equation of state.”

The equation of state is a set of empirically derived equations. When you set up a computer model for any process involving vapor-liquid equilibrium, you must select an equation of state from perhaps two dozen choices. Experience is the only real guide in making the most accurate selection.

When you set up a computer model for any distillation process, you must select a tray efficiency. Tray efficiencies (for mechanically intact trays) vary from 30 to 90 percent. Experience is the only guide in making the least wrong selection.

51.1.3 Establishing a Firm Design Basis

The design engineer for the new propylene splitter had used the following values in his model:

- Relative volatility = 1.18
- Tray efficiency = 65 percent

I had used in my computer model:

- Relative volatility = 1.21
- Tray efficiency = 75 percent

I had arbitrarily manipulated tray efficiency and relative volatility to force my computer model to match the observed plant data. It might seem that by arbitrarily selecting both the relative volatility and tray efficiency for my computer model, my calculations would be little better than a guess.

My computer model was indeed a guess, but an educated guess. It's an extrapolation from a known data point—that is, the observed operation of the existing splitter at a feed rate of 12,000 BSD and a reflux rate of 24,000 BSD.

That's it! Computer modeling in the process industry is at best a very approximate extrapolation technique. The model may begin to reflect reality insofar as it is based on a wide range of accurate plant data.

This suggests that an experienced process design engineer can design a plant based on rules of thumb, prototypes, and experience better than the novice engineer backed up by the latest in computer technology.

51.1.4 A Monument Rises

I forgot to mention something. I forgot to mention that the foundation for the new propane-propylene splitter had already been constructed. A meeting was held to review my report regarding the design of the new splitter. My comment that the existing tower was adequate for the projected 12,000 BSD feed rate was not well received. My recommendation that the new tower not be built was greeted with open hostility. My client's design engineer explained, "Mr. Lieberman, data such as your one-day plant test imply the existing tower is adequate. But the bulk of the available information, based on two years of computer simulations, has established that a new tower is required."

"Furthermore, Mr. Lieberman, we are terminating your contract immediately. We never asked you to look into the operation of the existing splitter. Your entire attitude has been politically incorrect."

Well, the new tower was built, a monument to ancient Greek philosophy, "Facts don't count."

51.2 Computer Control

On-line, closed-loop computer control frequently does more harm than good. This is not a popular opinion, but it is my opinion. Perhaps my observations only apply to refinery process units, rather than chemical plants in general. But I know what I've seen.

- Using computer control as an alternate to training panel board operators in the true nature of process control of their unit does not work. Reason: When a key process parameter changes, it is not recognized by the operators.
- Using computer control is fine, as long as the technician who developed and implemented the control strategy is still around. Once the computer control ascends to the status of a "Black Box," it becomes impossible to adjust the operating parameters to accommodate a changed situation.
- Restricting the authority of the panel board operator to override the computer control leads to a gradual loss of knowledge. With time, the panel board operator will fail to understand the interrelationship of the various operating parameters.

Then, when I come into the control room and note the reflux rate is falling when it should be rising, I'll hear, "The computer handles the reflux rate. It's not my responsibility." This is abusive use of computer technology. The operator has been turned into an observer and is no longer actively controlling the process.

- The best operators should not be used full time on the panel. This happens because the computer control has become too complex for the average operator. The best operators need to spend half their time outside on the unit.

I have often been persuaded that computer control is really working very well on a particular process unit. But more often than not, a closer investigation reveals the truth of what I learned in 1968. An experienced, intelligent, committed operator, fully involved in his or her duties, will outperform computer control on most occasions.

Mr. Shinsky of Foxboro, a famous expert in process control, once explained to me that most control problems are really process limits. Mr. Shinsky said that for computer control to succeed, we must update the physical plant limitation data. Nothing has changed in 30 years to make his wise words less true.

51.3 Material Balance Problems in Computer Modeling

Many computer models fail to represent reality because of faulty input of feed compositions. Common causes of wrong input of feed compositions are:

- Liquid stream compositions that do not reflect the lighter components in the sample. A sample of gasoline is collected in a bottle. The methane and half the ethane are flashed off as the bottle warms on the laboratory bench.
- Vapor stream compositions that do not reflect the heavier components in the sample. Half the pentane and most of the hexane condense out in the gas sample bladder, as the bladder cools waiting for the chemist to run the gas through the chromatograph.

One of my friends, Steve, made such an error in designing a debutanizer. He failed to account for the methane and ethane in the existing debutanizer feed stream. The sample from the 25-psig feed vessel was taken in a bottle. The lighter components weathered off prior to lab analysis. The results were, for Steve, rather catastrophic. The debutanizer overhead product could not be fully condensed. The non-condensable vapor pressured up the overhead reflux drum. The non-condensables were vented, along with 30 percent of the butane, to the flare. Steve was

promoted to manager of employee relations, an important but potentially less critical position.

51.3.1 How Steve Could Have Avoided becoming Employee Relations Manager

1. Obtain a gas chromatographic analysis of the debutanizer feed in mole percent.
2. Multiply the mole percent of each component by the vapor pressure of that component at the temperature in the feed vessel at the time the sample was drawn. This is the partial pressure of each component.
3. Add up the sum of the partial pressures to obtain the calculated total vapor pressure. Compare this calculated vapor pressure (in psia) to the actual pressure in the feed vessel (i.e., 25 psig or 40 psia).
4. If the calculated vapor pressure is close to the observed pressure, all is well. If the calculated vapor pressure is well below the observed pressure, proceed to step 5.
5. Get a fresh sample from the feed vessel. Heat it up by a few degrees—just enough to fill a gas bladder with the evolved gas. Send this gas bladder to the lab for gas chrome analysis.
6. Take the mole percent liquid composition obtained in step 1. Using the composition obtained in step 5 as a guide, manually add in the lighter components to the calculated mole percent liquid composition, until the calculated vapor pressure equals the observed pressure in the feed vessel.

51.3.2 Missing Heavier Components from a Vapor Sample

Steve compounded his error in the design of a sponge oil absorber used to recover heavier components from a waste gas stream. In this case, my friend failed to account for the large amount of isopentane that was recoverable from the methane and ethane waste gas. To avoid this error, Steve should have:

1. Submit a sample of the waste gas for analysis to obtain a mole percent composition.
2. Divide the mole percent of each component by the vapor pressure of the pure component at the temperature of the drum from which the sample was taken.
3. Sum up the above numbers. If the total is close to the reciprocal of the drum pressure (in psia) from which the sample was taken, all is well. If this sum is less than the reciprocal of the drum pressure, proceed to step 4.

4. Get a fresh bladder sample of the vapor from the drum. Condense out some of the vapor to liquid with cold water or ice. Obtain a gas chrome analysis of this liquid.
5. Take the mole percent vapor composition obtained in step 1. Using the composition obtained in step 4 as a guide, manually add in the heavier components to the mole percent vapor composition until the calculated sum of the:

Mole percent divided by vapor pressure

equals the reciprocal of the total absolute pressure in the drum from which the sample was taken.

51.3.3 The Garbage Rule

All I'm doing is checking the validity of my material balance input data. I'm using bubble-point and dew-point calculations to validate that my material balance represents reality. In the rush to create our computer models for process equipment, we may sometimes forget the fundamental rule of computer technology:

GARBAGE IN = GARBAGE OUT

If you don't believe me, ask Steve, now a well-paid but bored vice president of human resources for a well-known Texas corporation with headquarters in Houston.

51.4 Fourth Edition Update Comments

I wrote this chapter in 2002. Now, as I review computer control and computer modeling for this fourth edition in 2014, my attitude toward both aspects of computer technology has not been altered:

- Computer modeling must be based on an operating process system in similar service.
- Computer control only works in the long run if the control technician understands how to run his or her controls in a manual operating mode, and occasionally practice such manual control.

In the 700 plus process troubleshooting seminars I have instructed in the past 31 years, no experienced plant operator or design engineer has taken issue with the above two points.

CHAPTER 52

Field Troubleshooting Process Problems

Humans have instinctive drives, among which is the desire to fix things. I'd bet, having read this far, that you have that inner desire to make repairs.

Growing inside me, growing stronger as I age, is a frantic, fanatical need to repair mechanical devices. There are times when I've risked my life on a rickety scaffold to measure an outlet temperature from a heat exchanger.

My career is studded with the quick-fix solution to process problems. The quick fix that lost me a \$100,000 engineering fee to redesign a fractionation tower. That overpowering desire to impose my will on the inanimate world, regardless of the consequences.

52.1 De-ethanizer Flooding

- Place: Texaco Refinery, Convent, Louisiana
- Time: 11:45 P.M., December 24, 2001
- Weather: Cold, windy, rainy
- Problem: Premature de-ethanizer flooding
- Contract value: \$240,000 (U.S.)

Liz, Jerry, Mike, and I were partners in expanding the capacity of a naphtha reformer plant producing aromatic base stocks. The bottleneck was the reboiled de-ethanizer, limited by flooding. Our job was to produce a process design to de-bottleneck the reformer from

40,000 BSD (barrels per day) up to 50,000 BSD of feed. But I was stymied by the following:

- The calculated percent of jet or vapor flood was 65 percent at current feed rates (40,000 BSD).
- The calculated percent of downcomer or liquid flood was 70 percent at 40,000 BSD.
- The de-ethanizer was known to flood at 41,000 BSD feed.
- The tower had been opened twice before, found to be both clean and in good mechanical condition.
- In theory, the tower should be adequate at 50,000 BSD without any modifications.

I didn't know what to do. But fate intervened. A tube leak on an upstream heat exchanger necessitated a short unit outage. The de-ethanizer tower was opened for one day so that we could carry out an internal inspection of the trays.

I crawled through the tower with Mike. Cold washwater was still cascading down the tray decks. The trays were old-style valve trays. All 30 trays had a 0.5-inch valve cap lift (see Fig. 52.1). All 30 trays except, I found, the bottom tray, which had only a 0.25-inch valve cap lift.

Now my calculated ΔP dry (i.e., the pressure drop of the vapor flowing through the opening between the tray deck and the valve cap) was 2 inches of liquid with the 0.5-inch lift. With a 0.25-inch valve lift, the vapor velocity would double. The ΔP dry would increase with the square of the velocity to 8 inches. Further, the height of clear liquid in the downcomer would increase by an extra 6 inches of liquid. This would certainly flood the downcomer from the bottom tray. (See Chap. 3.)

Here, then, was my naphtha reformer bottleneck—too small a valve cap lift in the bottom tray of the de-ethanizer tower.

52.1.1 What's Next?

The wind was brutal as Mike and I, soaked and bruised, crawled out of the manway.

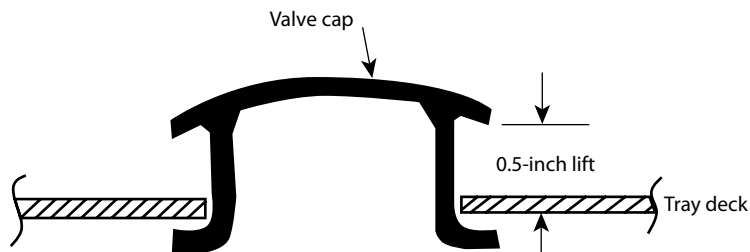


FIGURE 52.1 A valve cap, fully open, with 0.5-inch cup lift. Smaller lift promotes flooding.

"Well," asked the night mechanical foreman, "okay if I close the manways? It's 1:30 A.M. My guys are out here on overtime pay."

"No," I answered. "Pull out the bottom tray panels. I want you to drill sieve holes between the valve caps. I'll tell you how many sieve holes I want in 20 minutes."

I explained my plan to Liz and Jerry. It was too late to find new valve caps with a 0.5-inch lift. The next best thing was to increase the bottom tray open area with sieve holes to match the open area of the other 29 tray decks. Sixty 0.75-inch sieve holes would be drilled on the bottom tray panels.

Jerry and Liz agreed with enthusiasm. Mike, however, was horrified.

"Norm, what's all this about?" asked Mike.

"Look, Mike, I know what I'm doing. Have some confidence in me."

"Norm," Mike whispered, "one of two things are going to happen with your stupid sieve holes. Either your idea will work, or it won't work."

"Get to the point, Mike, I'm cold and getting colder."

"The point is, if your damn sieve holes don't work, Texaco will think we don't know what we're doing and we'll lose the contract. The point is, if your sieve hole idea does work, Texaco will cancel the project because the bottleneck will be gone. Either way we are going to lose \$200,000. That's my point," Mike screamed.

"Kindly do not yell at me," I answered calmly.

"Okay, Norm, then kindly explain to me your purpose for drilling the sieve holes."

"Mike, if you don't know, I can't explain it."

The sieve holes were drilled, the plant bottleneck was eliminated, we lost the contract, Mike is still angry, Texaco is now Motiva. But I'm happy. But if you don't know why, dear reader, I can't explain it.

52.2 The Elements of Troubleshooting

I once found a blue sapphire in Australia. It was imbedded in a gravel bed. It had been buried after a volcanic explosion in the gravel of the old stream bed for 65 million years. It was easy to find. All I had to do was to dig up and sift through 4 tons of gravel. It only took six days. But I knew it was there, so it was easy to find.

Troubleshooting is like that. Part of every problem, embedded in the problem itself, is the solution. The solution is a component of the problem. Each problem consists of a number of components. And one of these components is the answer you seek.

The first element of troubleshooting is an absolute belief that the solution exists. An absolute belief that you can find the answer.

An absolute belief that it's only a matter of *time* and *determination*. An understanding that you have to do the job alone. No one can help you because no one else is dedicated enough. No one else cares enough.

The second element of troubleshooting is direct field observations. There is a piece of data you lack to intellectually grasp the problem. The difficulty is you don't know what that piece of data is. It's like looking for gems in the Australian outback. Often, a client asks me why I so desperately wanted some particular pressure. My answer is that it was physically possible to make the pressure measurement. I want all the data available because I don't know beforehand which piece of data I really need. So I want all of it. How can you tell what samples, temperatures, and pressures are available if you don't go out to the field and look?

My personal preference is to obtain all plant data using my own tools, wrenches, gauges, temperature probes, and fittings myself. In my toolbox, I have every 0.25- to 1.5-inch pipe fitting known to mankind. I also have a set of instrument fittings and every possible adapter to convert from piping threads to instrument threads.

I routinely drag 30 pounds of bushings, elbows, sleeves, and nipples all over the world. I never know when I'll need a 1- to 0.5-inch reducer sleeve to make that definitive pressure measurement.

But why do I have to unscrew that piping plug myself? Why must I use my own hands to pipe-up that pressure gauge? Why, at 72 years old, do I insist on climbing up 120 feet to a tower vapor line?

First, the physical activity relaxes me. Also, it gets me away from other people. What I need is time to reflect on the problem without distraction. Also, if I get the data myself, I know they are correct. Or at least I know the range of uncertainty of the data.

Basically, I just like climbing tall distillation towers and working with pipe wrenches and fittings. It's fun. I guess one element of troubleshooting is to think it's all a game, a challenge to be enjoyed. Honestly, I'd do it for free.

52.3 Field Calculations

Some people use calculators or computers. I have my K&E slide rule. I used to be a computer guy in the days of punch cards and Fortran. But now I always take an electrochemical computer with me on all field troubleshooting assignments. It fits compactly beneath my hard hat.

The ability to manipulate field data as it is obtained is the best way to speed success in field troubleshooting. For example, how much air is leaking into the incinerator duct at the sulfur plant in Aruba? What size hole in the duct is needed to allow this much air leakage? I can calculate the air flow based on the observed temperature rise in the duct due to afterburning of hydrogen sulfide in the incinerator effluent. Now I know that I need to find, and did find just yesterday, a 4-inch by 4-inch hole in the duct.

What you need to do is carry into the field:

- Your technical training
- The tools of the trade
- The fanatical desire to solve the problem
- The absolute certainty that the solution exists and is accessible to you

Most of the equations you will need for common process equipment have been presented in the preceding chapters. One of the purposes that we had in writing this book was to make such information available for field troubleshooting.

52.4 Troubleshooting Tools—Your Wrench

The tools of my trade are:

- Gear wrench. This is a two-headed wrench. One end is a valve wrench for turning valve handles. The other end has adjustable jaws for piping and fittings.
- 6-inch crescent wrench for instrument fittings.
- 12-inch crescent wrench for pressure gauges.
- A 16-inch aluminum adjustable pipe wrench (steel is cheaper but twice as heavy).
- A short, curved knife for cutting through insulation.
- A digital pressure gauge that reads both vacuum and pressures to 30 psig.
- An infrared temperature gun that reads up to 800°F.
- An infrared temperature probe with a laser pointer that reads up to 500°F.
- Spare, 9-V batteries for the last three items.
- A hand-powered vacuum pump used for obtaining flue gas and samples of gas under vacuum down to 25 inches of Hg. This is a \$30 item purchased in an auto supply parts store. Its conventional use is to evacuate brake fluid lines.
- Plastic tubing.
- 5 ft of coiled stainless steel 0.25-inch tubing, used as a sample cooler.
- Litmus paper for pH.
- Lead acetate paper to check for hydrogen sulfide.
- Inflatable plastic bag to catch gas samples.

- About 60 assorted piping fittings and 10 assorted instrument fittings.
- A section of 0.5-inch stainless tubing with an instrument fitting at one end and a piping fitting on the other end.

52.5 Field Measurements

My success as a process engineer is in proportion to my ability to take accurate field measurements, most often to obtain a differential pressure between two points. Since the differential pressure is usually only a small portion of the total system pressure, consistent, if not precise measurement is critical.

Use a digital pressure gauge of the smallest range. If you are checking pressures on a distillation tower that operates at 80 psig, use a 0- to 100-psig gauge, not a 0- to 500-psig gauge. The larger pressure range gauge has a potential error five times greater than the smaller range gauge. Digital gauges are always preferred because everyone reads the same pressure. Also, the orientation of the gauge does not affect its reading. A good-quality digital gauge will cost \$500 to \$1000 (U.S.). Radio transmissions affect the digital gauge reading, so shut off your radio when taking such a reading.

Velocity affects pressure. Figure 52.2 illustrates the problem. This is plant data from a hydro-desulfurization unit. The 10-psi increase in the pressure through the exchanger is due to the reduction in velocity, resulting from flowing from the 6-inch exchanger inlet into the 8-inch exchanger outlet line. Certainly there are frictional losses through the exchanger, but this is more than offset by the conversion of kinetic energy into pressure.

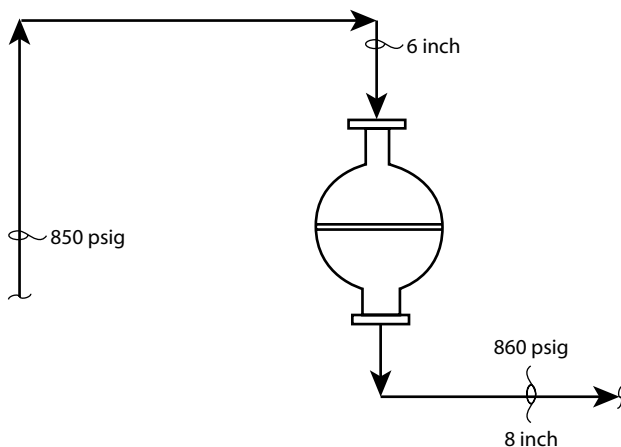


FIGURE 52.2 Decrease in velocity increases flowing pressure by 10 psig.

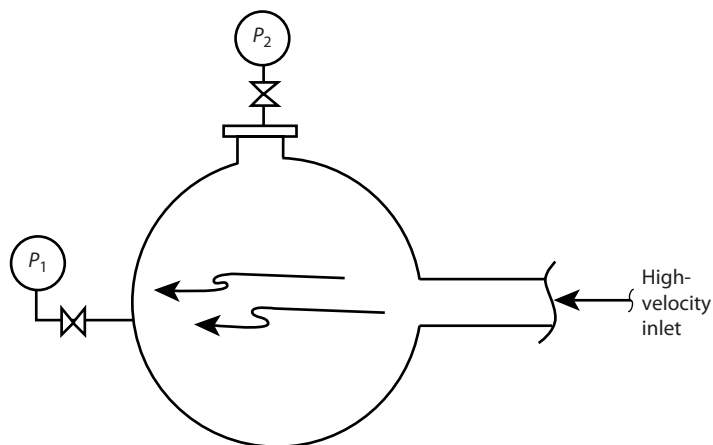


FIGURE 52.3 P_1 will read higher than P_2 , due to dynamic pressure—the conversion of velocity into localized pressure.

When measuring a differential pressure, changes in elevation between the two pressure points should be corrected to take into account the density of the flowing fluid (one psi equals 2.31 ft of water). (See Chap. 48 for an example.)

Avoid “dynamic pressure” readings. Figure 52.3 illustrates the problem. The kinetic energy of the high-velocity inlet fluid will be converted to pressure as it impacts the vessel wall at P_1 . This is called a dynamic pressure. The pressure at P_2 will be lower than P_1 . The correct pressure to read is the static pressure measured at P_2 .

Account for nozzle entrance and exit losses. (See Fig. 52.4.) The pressure at P_2 is lower than P_1 due to the acceleration of vapor through the overhead vapor line. That is, the pressure of the vapor is partly converted to velocity. To correct the pressure at P_2 to static pressure at P_1 , add the nozzle exit loss as follows:

$$\text{Nozzle exit loss (psi)} = 0.3 \times \frac{(DV)}{(62.3)} \times \frac{V^2}{(27.7)}$$

where DV = density of vapor in lb/ft³

V = vapor velocity in overhead line in ft/s

Better yet, just measure the pressure at P_1 and forget about the corrections for nozzle losses. On the other hand, the fluid flowing from a nozzle into a vessel will usually gain in pressure (pressure recovery).

Don't forget about time. Pressures in flowing systems naturally fluctuate by a few percent. To avoid the distortion of an observed differential pressure due to such fluctuations, select a reference pressure somewhere in the process. After each pressure measurement, recheck

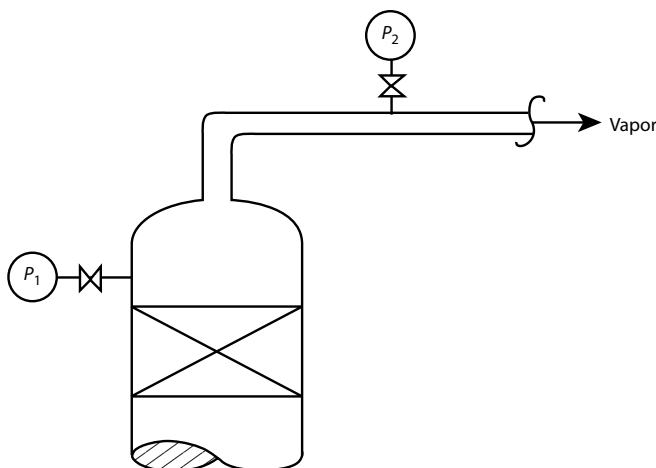


FIGURE 52.4 Pressure at P_2 is lower than P_1 , due to nozzle exit loss. Pressure at P_1 is converted to velocity at P_2 .

the reference pressure gauge. Correct your recent measurements for the observed pressure fluctuations.

I could write a whole book just about making pressure and flow and level and temperature measurements in the field. Actually, I have. (*Troubleshooting Process Operations*, 3d ed., Pennwell Publications, Tulsa, Oklahoma.) But I've made my point. Making field measurements is complicated. There are too many variables to turn the job over to the outside unit operator. There are so many sources of potential error to consider and eliminate. There is only one way to make sure any job is done right and that's to do it yourself. At least you will know which data are reliable and which data are questionable.

52.5.1 Indirect Flow Measurement

Relief valves, control valves, and isolation valves are subject to leakage. In a typical modern petroleum refinery, about 0.4 wt.% of crude charge is sacrificed in this wasteful manner, mainly due to valve leakage. Knowing that a valve is leaking is not much help in getting it fixed, unless an approximate idea can be obtained as to how much is leaking. Let me give an example as to how I handle this problem.

Assume that a closed gate valve on a 3-inch, 200°F, hot water line is leaking. We are sure that the valve is leaking because the piping downstream of the closed gate valve is hot to the touch. To estimate the leaking flow rate, proceed as follows:

1. Using your surface temperature infrared gun, check the temperature of the pipe immediately downstream of the leaking valve.
2. Repeat the above, 20 ft downstream.

3. Calculate the observed sensible heat loss per pound of water. If the observed temperature difference between step 1 and step 2 is 20°F, then heat loss per pound is 20 Btu per pound of water.
4. Calculate the observed ambient heat loss from the 20 ft of 3-inch pipe (with an area of 16 ft²) using the formula:

$$\text{Heat loss} = (\Delta T) (U) \text{ area}$$

where:

- Heat loss = Btu per hour.
- ΔT = Temperature difference between the average pipe surface temperature and the ambient air temperature, °F.
- U = Heat transfer coefficient. Use about 2 for skin temperatures of 200 to 300°F in still air. Use about 4 for skin temperatures of 400 to 600°F in windy conditions.
- Area of pipe, ft².

For our example:

- Heat loss = (180 to 80°F) (2) (6) = 3200 Btu/hr.
 - The (180 to 80°F) term is based on the average skin temperature of the pipe of 180°F and the ambient temperature of 80°F.
5. Divide the observed ambient heat loss (step 4) by the observed sensible heat loss per pound of water. In our example:
 - 3200 Btu/hr ÷ 20 Btu/lb = 160 lb/hr

This is the amount of 200°F water leaking through the closed gate.

If the pipe is insulated, just make the skin temperature measurements on the external insulation. The heat transfer coefficients (between 2 and 4) will still apply in the temperature ranges noted above.

52.5.2 Using the Infrared Surface Temperature Gun

Checking a pipe surface temperature using a temperature gun requires following several rules:

- The gun needs to be 1 to 6 inches from the surface. Do not be fooled by the laser pointer. Just having the red dot on the surface will not give you a good reading on a small pipe if you are 5 ft away. If it's a large vessel, the longer distance is okay.
- A shiny reflective surface will give erratically low readings. Stainless steel surfaces are bad. Rub some tar or black paint on such a surface. Wait until it stops smoking. A rough, rusty,

black, or brown surface is best. For cool (100°F) pipes, surface conditions are not important.

- Try for the highest temperature reading possible. The angle you hold the gun at, relative to the pipe, does affect the observed temperature.
- Don't waste time and money on an instrument with an emissivity reading. Just make sure the pipe surface is nonreflective by using the tar, black paint, or blackish grease. I've had no luck with the emissivity settings.

Use of the surface temperature gun, which is my main troubleshooting tool, requires consistency, not accuracy.

52.5.3 Pressure Measurement Problem

Often one has to obtain a pressure reading on a vapor-filled line. Most technicians will realize that locating the gauge below the vapor line will result in an erroneously high reading because the line will fill with condensate. But suppose the pressure gauge is located above the vapor line. If the line is self-draining (no loops or pockets), will a true pressure reading result? It depends on the diameter and length of the pressure sensing line. A sensing line less than 10 inches is fine. If the pressure sensing line is 2 inches or larger, it will be self-draining and that is good. But if the pressure sensing line is smaller than 0.75 inch, the line will not be self-draining. The condensate will accumulate in the vertical section of the line, and the indicated pressure reading will be too low. Why this happens I cannot say. But I have observed this problem for both hydrocarbon and aqueous systems.

52.6 Troubleshooting Methods

With tool bag in hand, you now set off to troubleshoot the process problem. There are three alternative methods to consider:

- Experience
- Trial and error
- Insight

To a large extent, experience does not help me solve very many problems. It isn't that I don't see the same problem repeatedly. I certainly do. It's just that a problem has so many different faces that I fail to recognize the true nature of the problem. Superficially, it looks unique.

Relying on experience wastes time. Relying on experience as a guide drags me into the quagmire of trial and error. I have an inner weakness for trial-and-error solutions. It's inertia and laziness on my part. It's part of my propensity, carried forward from childhood, for the quick fix without the commensurate effort.

It's true many problems are solved by experience (someone has seen the same problem before). It's true most problems are solved by trial and error (alternatives are tried until something works). My difficulty is that process problems that can be solved by trial and error or by ordinary experience rarely reach me. They are handled by the shift operators or solved by the unit engineers.

It's the intractable problems that are my domain. The ones caused by several malfunctions, all of which have to be corrected before any improvement is obtained. It's the problems that appear to be unrelated to any prior plant operating experience.

These sorts of problems I have been able to solve only by insight. First, I collect all the field data. Second, I discuss the situation with the plant shift operators. Next, I'll examine tower internal drawings, pump curves, and exchanger data sheets. Finally, I'll perform the relevant engineering calculations. This usually does not result in reaching a resolution of the problem.

What I must now do is that which I find most difficult. It's the application of intellectual energy and insight.

I have to be alone. Alone and away from telephones. I do best when out of the plant, perhaps walking in a park.

I have to accept all the data. I'll repeat to myself the mantra, "Even though you don't like the data, the data represent reality. The data you wish to reject are just a reflection of your lack of intellectual honesty."

I have to put aside all preconceptions. I have to discard the notion that I've seen this situation before. Trying to solve a problem based on experience creates mental confusion and wastes time.

I have to cast aside my ego. Often I've already taken a position in some meeting. I hate to admit I'm wrong. I hate to admit that an analysis by some junior engineer was correct.

Thirty-five years ago a young engineer from Standard Oil of Ohio asked me to define a step wise approach to troubleshooting process plant problems. Honestly, I didn't know how to answer her question. And I still don't know. It's a sense that grows inside me, rather like hunger or anger or joy. It's a sense of growing comprehension. The seed of the solution existed in my mind all along. It will grow into comprehension given the right environment.

I call this insight, but if you don't know what I mean, I can't explain it.

52.7 Afterword

I started work as a process engineer at the long since vanished American Oil Refinery in Whiting, Indiana, on November 19, 1965. William Duval, my boss, handed me a large, loose-leaf manual entitled "Standard Design Practices." What seemed odd at the time was the archaic contents of the manual. Most of the sections were dated before World War II, early in the 1930s. The section on knockout drum sizing hailed from the mid-1920s.

I left my job with American Oil in 1981 and stole their “Standard Design Practices” manual. I can see its faded blue binder now. Not in my imagination. It’s on my bookshelf with my other reference texts. Just feet from where I’m sitting. I use it still when I design process equipment and evaluate malfunctions. Including the vapor-liquid separation chart from the 1920s.

We work in a stagnant industry. It’s true that vapor-liquid separation can be greatly improved by the use of vertical vortex tubes, which are a relatively new development. Yet most vapor-liquid separators are still empty vessels that depend on gravity settling and are sized in accordance with Stokes’ Law.

We have distillation trays using “Super-Frac” advanced technology. Yet bubble-cap trays, developed in the 1830s, are in widespread use, and refiners still rely on sieve trays.

There have been tremendous advances in the design of centrifugal pump mechanical seals, yet many new pumps employ ancient rope-type packing to prevent product leakage around the shaft.

Computer control has transformed the design of control systems in the process industry. Yet the panel operator must still be trained to run the plant manually, using 1920s control loops when the computer control falters, and even then must also be well practiced at manual control under daily normal circumstances in order to cope with plant upsets.

Helical baffles, tube inserts, and twisted tubes are all becoming more common in new heat exchangers. Yet 99 percent of the shell-and-tube heat exchangers Liz and I encounter are of conventional design.

So, if process equipment itself has not changed, why this fourth edition?

Well, something has changed. I’ve changed. Since Liz and I authored the third edition, we have both become smarter. Smarter in the sense that we have learned a lot about our misconceptions about how process equipment works. Much of our understanding about vacuum ejectors, centrifugal compressors, pump mechanical seals, and distillation tray hydraulics were not correct. We’re both learning all the time. So we must be getting smarter.

Maybe if we keep on learning from our mistakes, Liz and I can get smart enough to stop making errors. Wisdom comes with age. As long as we both remember that we are not too old to learn, we’ll be okay.

May 7, 2014

Thanks for reading our book. If you have any questions, please contact us at:

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Glossary

- Actual level** This is the liquid level actually in the vessel.
- Adiabatic combustion** Burning fuel with no radiant heat losses.
- Adiabatic process** No heat lost or gained.
- Aeration factor** The actual density of a froth, divided by its clear liquid density.
- Affinity or fan laws** Relates speed of centrifugal machines to head, flow, and work.
- Afterburn** Re-ignition of flue gas in the convective section of a fired heater.
- Air to open** Control valves are operated by air pressure. A control valve that is opened by air pressure is “air to open.”
- Apparent level** This is the liquid level that appears in the gauge glass or is displayed on the control panel.
- Available NPSH** The difference between the pressure at the suction of a pump and the vapor pressure of the liquid.
- Barometric condenser** Condensation under vacuum by direct contact with the cooling water.
- Blown condensate seal** Vapor or steam flowing through the condensate outlet nozzle of a condenser.
- Boiler codes** Laws describing the safe design for boilers and other pressure vessels.
- Boiler feedwater** Water that has been softened or demineralized and deaerated.
- Bottom dead center** The piston position in a cylinder when it is closest to the crank shaft.
- Carbonic acid** Corrosive CO_2 dissolved in water.
- Cavitation** Flashing of liquid in a confined space, such as a nozzle.

Centipoise The most common measure of viscosity.

Centistokes A measure of viscosity.

Chain-locked open The practice of passing a chain through a valve handle and locking it in place, to prevent it from being closed without the proper authority.

Channel head The portion of a shell-and-tube heat exchanger that covers the channel head tubesheet.

Channel head cover A round plate that bolts onto the channel head.

Channel head pass partition The baffles that divide a heat-exchanger channel head to create multipass exchangers.

Channel head tubesheet The fixed end of the tube bundle.

Channeling Loss of contacting efficiency due to uneven flow distribution.

Clear liquid Density of liquid assuming no foam.

Cogeneration Use of turbine exhaust steam to provide process heat.

Compression ratio The discharge pressure divided by the suction pressure.

Compression work Energy provided to a gas to increase its pressure.

Condensate drum balance line Used to connect the channel head of a steam reboiler to the condensate drum for pressure balance.

Condensing turbine Exhausts into a surface or barometric condenser running at a vacuum.

Convective heat The sensible-heat content change of a flowing fluid.

Corrosion allowance Amount of extra metal thickness added to a pipe or vessel wall over and above that required for pressure to allow for future corrosion.

Crank end The end of the cylinder closest to the crank shaft.

Critical flow Vapor flowing at the speed of sound.

Critical speed The natural harmonic frequency of a centrifugal compressor, which determines its critical or self-destructive speed.

Cross-flow velocity Fluid flow at right angles to a heat-transfer tube.

Deaerator Used to remove oxygen from boiler feedwater.

Degree of fractionation A measure of how much of the light components are left in the bottoms product, and how much of the heavier components are left in the overhead product.

Delta P Used as shorthand for pressure drop.

Demister pad A steel, plastic, or straw mesh pad used to coalesce droplets of liquid.

Dependent variables Parameters that an operator does not have direct control of.

Dew-point equation Used to predict when a vapor will start to condense.

Downcomer seal On a distillation tray, the height of liquid above the bottom edge of the downcomer.

Draw-off boot A low point in a vessel used to collect water.

Drip points On a packed bed, this refers to the number of points where liquid enters the top of the bed.

Driver-limited A pump that is limited by the horsepower of its turbine or motor.

Drying out Liquid flowing down trays in a tower progressively evaporating.

Dynamic machines Pumps or compressors that convert velocity into feet of head.

Elevation head loss When liquid flows uphill, it loses pressure due to the increased elevation.

Emulsion A mixture of two insoluble phases, such as oil and water.

Entrainment coefficient A measure of the tendency of a flowing gas to carry droplets of liquid.

Entropy A measure of a fluid's ability to do work.

Equilibrium conditions Vapor and liquid in intimate contact are said to be in equilibrium.

Excess O₂ A measure of the amount of excess air used in combustion.

Expansion joint A flexible section of piping with bellows, which permits thermal expansion.

Explosive region The ratio of fuel to oxygen that will sustain combustion.

Extended surface Fins, studs, or serrations on the outside surface of heat-exchanger tubes.

Eye of the impeller The inlet nozzle in a centrifugal pump's impeller.

Fan blade pitch The angle of a fan blade. The bigger the pitch, the greater the air flow.

Fan laws Relationship between speed, head, and flow for centrifugal machines.

Feed preheater The heat exchanger used to increase the enthalpy of the feed to a distillation tower.

Film resistance The property of a fluid that diminishes heat exchange.

Fin tube bundle Serrated, or extended-surface, exchanger tubes.

FLA Full-limit amps, or the high-amperage trip point for a motor.

Flange rating Piping and heat-exchanger flanges have a nominal pressure rating less than their actual pressure rating.

Flash points Temperature at which a liquid will ignite when exposed to a flame.

Floating head The component of a shell-and-tube exchanger that covers the floating-head tubesheet.

Floating-head exchanger Shell-and-tube exchanger in which one end of the tube bundle is left free to move to accommodate thermal expansion.

Floating-head tubesheet The end of a tube bundle left free to move, due to thermal expansion.

Flooded condenser control A method of pressure control in a distillation tower.

Flooding In distillation, the point in the operation where increases in vapor-liquid loading cause a noticeable reduction in the fractionation efficiency or tray efficiency of the tower.

Forced circulation Circulation in reboilers that receive their liquid feed from the discharge of a pump.

Frictional loss Pressure reduction accompanied by heat generated by a fluid flowing through piping.

Gauge glass Used to measure the pressure difference between two points in a vessel in terms of the density of liquid in the glass.

Governor speed-control valve Controls a turbine's speed automatically by adjusting the steam flow into the steam chest.

Head end The end of the cylinder farthest away from the crank shaft.

Head loss Pressure lost due to friction in a flowing liquid.

Head pressure The pressure exerted by a height of liquid.

Header box A box from which a number of pipes emerge.

Heat balance Heat inputs have to equal heat outputs and ambient heat losses.

Heat flux Heat input per unit of time per unit of heat-transfer surface area.

Heat-proving the heater Optimizing air flow to maximize combustion efficiency.

Heat-stable salts Formed in an MEA regenerator.

Heat-transfer surface area The outside surface area of a tube used to transfer heat.

HETP Height equivalent to a theoretical plate (or tray). Also called an *equilibrium separation stage*.

High-temperature creep The plastic deformation of steel at high pressure and high temperature.

Hogging jet A single-stage jet exhausting to the atmosphere.

Holdup The time that liquid is retained in a vessel, usually to promote separation or provide hold time for a downstream pump.

Hot spot The glowing area on a heater tube.

Hot well The drum located below a surface condenser in vacuum service, where the barometric legs drain to.

Hydraulic hammer The correct term for water hammer.

Hydraulics The study of fluid flow, pressure, and feet of head.

Hydrogen activity The tendency of ionic hydrogen, or protons, to dissolve into the lattice structure of steel walls.

Hydrogen blister A result of corrosion leading to vessel failure.

Ideal compression work Compression work discounting friction, leakage, and other mechanical losses.

Ideal-gas law An equation describing the theoretical relationship of pressure, temperature, and volume of gas.

Impingement plate Used to protect tubes from the erosive velocity of the shell-side inlet fluid.

Incipient flood point The vapor-liquid loading at which a distillation tray develops its maximum efficiency.

Independent variables Parameters that an operator has complete control to change.

Indicator card A record of the pressure vs. volume of gas inside the cylinder of a reciprocating compressor.

Intake valves The inlet port valves of a reciprocating compressor.

Internal reflux The liquid flow leaving a distillation tower tray.

Irreversible process A machine that converts the pressure of a fluid into work always converts some of the potential work into heat. This heat is a measure of the irreversibility of the process.

Isoenthalpic An expansion of steam that does not convert heat to velocity. An irreversible process.

Isoentropic A process change where the ability of the system to do work has not been reduced. A reversible process.

Jet performance curve Issued by the steam jet manufacturer, describes the vacuum a jet should pull at various gas flow rates.

Kettle reboiler A type of reboiler that depends on the force of gravity for its liquid feed.

Laminar flow Fluid flow with little turbulence.

Latent heat Heat needed to change a liquid at its boiling point into a vapor at its dew point.

Latent-heat transfer Heat exchange involving vaporization or condensation only.

Letdown valve Used to reduce the pressure of a liquid refrigerant.

Lift gas Vapors used to lift solids or liquids to a higher elevation.

LMTD The natural log-mean temperature difference that promotes heat transfer.

Loop seal Used to prevent vapor from blowing back through a pipe to an area of lower pressure.

Mass flow Pounds of flow.

MEA reclaiming Used to remove heat-stable salts from amine.

Metering pump Used to both pump and measure the flow rate of small chemical additive streams.

Milliamp output The small (milliampere-range) electrical output from a thermocouple or pressure transducer that is used for control purposes.

Mils per year One mil equals 0.001 inch. Mils per year is a measure of how fast a steel wall corrodes away.

Molecular weight The weight of a gas divided by the number of moles of the gas.

Mollier diagram A chart relating enthalpy, entropy, temperature, and pressure of steam.

Motor pulley The wheel fixed to the shaft on a motor used to turn a belt drive.

Nozzle exit loss Conversion of pressure into velocity.

NPSH Net positive suction head.

Open area In a packed bed, the open area available for vapor flow.

Out-of-level The difference between the low point and high point of a distillation tower tray.

Overramping Tripping a motor because of excessive electrical power demand. Exceeding the FLA limit.

P&ID Piping and instrumentation diagram(s): the drawing(s) that most completely describe the process construction of the plant.

Partial pressure The vapor pressure of a component multiplied by its mole percent concentration.

Phase rule Fundamental concept relating the number of independent variables in a process.

Point of absolute combustion Air rate required to maximize heat recovery into the process, and minimize heat lost up the stack.

Polar molecule A molecule, such as water, that can conduct electricity.

Pressure recovery Conversion of velocity into pressure.

Pressure transducers A device that converts pressure into a small electrical flow.

Prime Filling the case of a pump with liquid.

Pulsation An acoustical phenomenon that causes a variable pressure inside the cylinder of a reciprocating compressor.

Pumparound A method to remove heat from a section of a tower.

Pumparound draw tray The tray from which liquid is drawn off to the pumparound pump.

Pumparound return tray The tray below the pumparound return nozzle.

Ratio of the phases A concept in level measurement, in which the relative heights of gas, hydrocarbons, and water must be considered.

Reboiler Source of heat input to a distillation tower.

Redistributor A plate that takes vapor or liquid flowing through a tower and, by means of orifices, forces the fluid to flow more evenly through a packed bed.

Reflux comes from the reboiler Concept that the reflux flow originates from heat supplied from the reboiler.

Relative efficiency The efficiency of a compressor as compared to the efficiency of the same machine at a different time.

Relative volatility The ratio of the vapor pressure of the light component divided by the vapor pressure of the heavier component at a particular temperature.

Required NPSH The net positive suction head needed to keep a pump from cavitating.

Reversible process Work extracted from the velocity and pressure of a fluid, which can be put back into the fluid to restore its pressure and temperature. An isentropic process.

- Rotating assembly** The portion of a centrifugal machine that spins.
- Rotor** The spinning part of a centrifugal compressor or pump.
- Running NPSH** The required liquid head needed to convert into velocity as the liquid flows into a centrifugal pump.
- Saturated liquid** Liquid at its boiling point.
- Saturated steam** Steam at its dew point.
- Saturated vapor** Vapor at its dew point.
- Secondary ignition** Re-ignition of the flue gas in the convective section of a heater.
- Self-flushed pumps** A centrifugal pump that obtains its seal mechanical flush liquid from the pump's own discharge.
- Sensible heat** Heat content due to temperature.
- Sensible-heat transfer** Heat exchange without change of phase.
- Stab-in reboiler** A reboiler tube bundle inserted directly into the base of a tower.
- Starting NPSH** The required liquid head needed to overcome the inertia of static liquid in the suction of a centrifugal pump.
- Starting volumetric clearance** The space between the piston and cylinder header at the end of the stroke in a reciprocating compressor.
- Stator** The stationary component on a centrifugal compressor.
- Steam chest** The portion of a steam turbine between the governor inlet valve and the turbine case.
- Steam rack** Device that automatically opens and closes steam nozzles into a turbine case.
- Steam tables** A table summarizing the enthalpy, density, pressure, and temperature of steam.
- Steam trap** Used to drain water from a steam system.
- Stuttering feed interruption** Erratic flow of liquid into a heater.
- Subcooled** Liquid below its boiling point.
- Sublimation** The change of a solid directly to a vapor—or the reverse.
- Super-fractionation** Effect of partial condenser or once-through reboiler on fractionation.
- Surface condenser** A condenser having separate vapor and liquid outlets.
- Surge** An unstable and dangerous operating mode of a centrifugal compressor.

Temperature difference In distillation, the tower bottom temperature minus the tower top temperature.

Thermowell The casing used to protect the thermocouple wires.

Top dead center The position of a piston in a cylinder when it is closest to the cylinder head.

Top-tray flooding Tower flooding caused only by flooding on the top tray.

Total dissolved solids (TDS) A measure of the tendency of boiler feedwater to form scale on a boiler's tubes.

Tramp air Air accidentally introduced into a firebox, but not through the burners.

Tray spacing The vertical spacing (typically 24 inches) between tray decks in a distillation tower.

Tube pitch Space between the centers of adjacent tubes in a heat exchanger.

Tube skin temperature The exterior temperature of a heater tube.

Tube support baffles Used to support tubes in a shell-and-tube heat exchanger.

Turbine case The portion of the turbine containing the wheel.

Turndown The minimum percentage of the design capacity at which process equipment will operate and still maintain a reasonable degree of efficiency.

Valve disablers Used in reciprocating compressors to reduce capacity.

Valve ports Openings in a reciprocating compressor's cylinder that permit the entry or exit of the flowing gas.

Velocity boost Pressure increase in a steam jet due to the reduction of velocity.

Velocity steam Used to increase the velocity of a liquid in a heater tube.

Vertical baffle Used in the bottom of distillation towers to segregate the cold-and-hot zone for more efficient reboiling.

Volumetric efficiency A measure of the volume of gas moved per stroke in a reciprocating compressor.

Vortex breaker A mechanical device, set over an outlet nozzle, to prevent drawing of vapor out of the nozzle with the flowing liquid due to the formation of a vortex.

Waste-heat boiler Used to generate steam from excess process heat.

Water hammer The sudden velocity reduction of a flowing liquid is converted into pressure surges, which then produce water hammer.

Water rate Pounds of steam needed to generate 1 horsepower (hp) in a steam turbine.

Weir loading On a distillation tray, the GPM (gallons per minute) of liquid flow divided by the inches of outlet weir length.

Wetted surface area In a packed bed, the area of the packing exposed to the internal reflux. A measure of the packing's vapor-liquid contacting efficiency.

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Condensers

- Fundamentals of Condensation
- Back-Flushing & Acid Cleaning Condensers
- Pressure Control of Columns
- Subcooling Reduces Condenser Capacity
- Surface Condensers

Vacuum Towers

- Improving Vacuum Tower Operation
- Packing and Grids in Vacuum Towers
- Steam Jets
- Vacuum Tower Overhead Systems

Pumps

- Fundamentals of Centrifugal Pumps
- Optimizing Impeller Size
- Cavitation & NPSH
- Causes of Seal & Bearing Failures

Compression

- Fundamentals of Gas Compression
- Surge in Centrifugal Compressors
- Variable-Speed Centrifugal Compressors
- Steam Turbines
- Reciprocating Compressor Efficiency
- Compressor Motor Over-Amping

Treating

- Amine Regeneration & Scrubbing
- MEA Degradation
- Sulfur Plant Start-Up
- Sulfur Plant Pressure Drop
- Jet Fuel Treating

Refinery Processes

- Troubleshooting Sulfuric Acid Alky
- Crude Tower Overhead Corrosion
- HF Alky Pressure Problems
- Troubleshooting Delayed Cokers—Part I
- Troubleshooting Delayed Cokers—Part II

Safety

- Field Testing Alarms & Trips
- Failure of an Amine C3–C4 Scrubber
- Boiling Water Causes Coker Fatality
- Sulfur Plant Safety Hazards
- Sources of Autoignition

To Order

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